Typical blade pitch ~5+

Need 2 levels Hi speed 18
Lo 12

blade pitch
-25 to 20
~~30 to~~ 100°
0 open

density

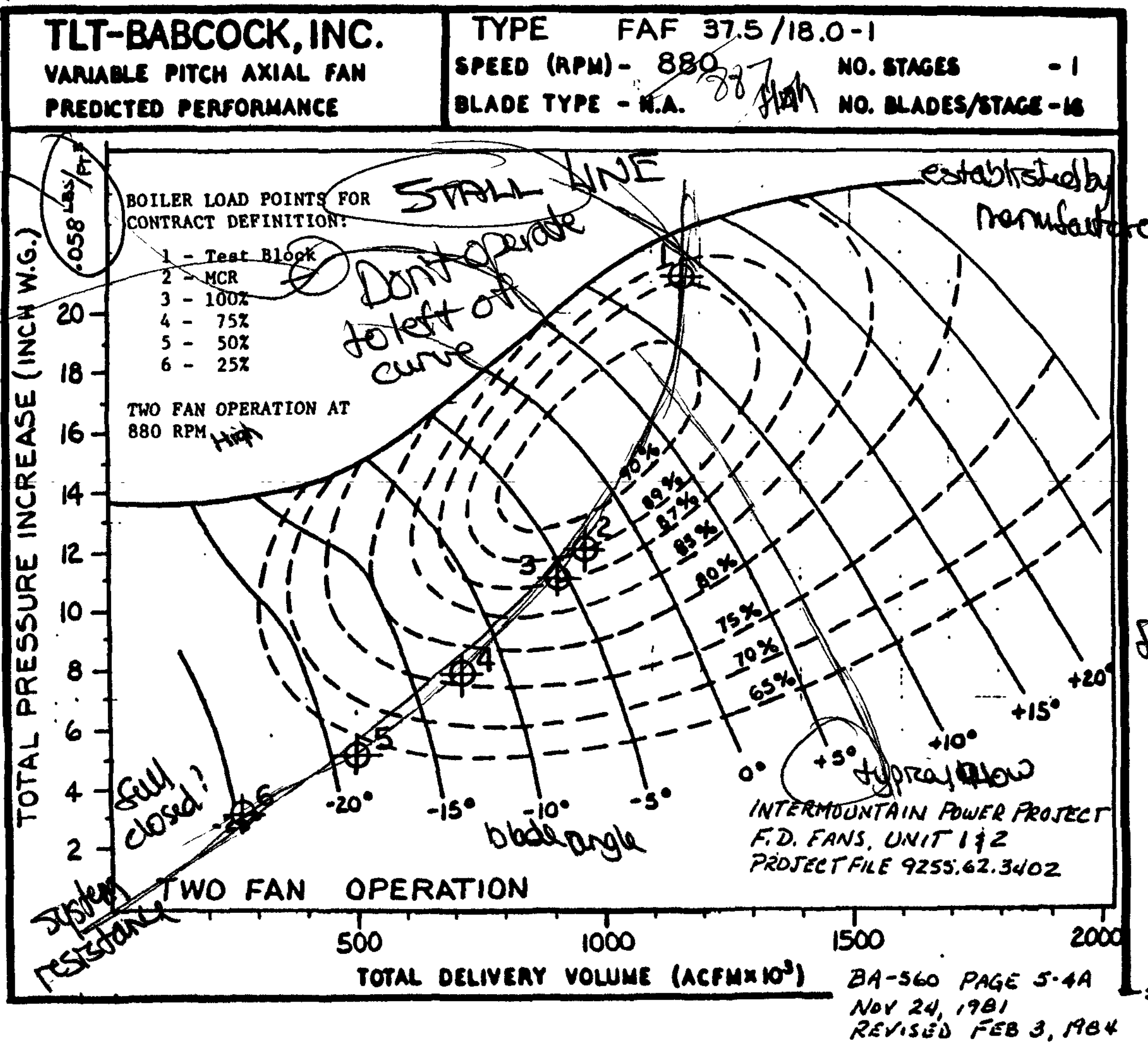
true stall area

Curves have been cut off

3-3

blade angle changes

typical disc ~5°

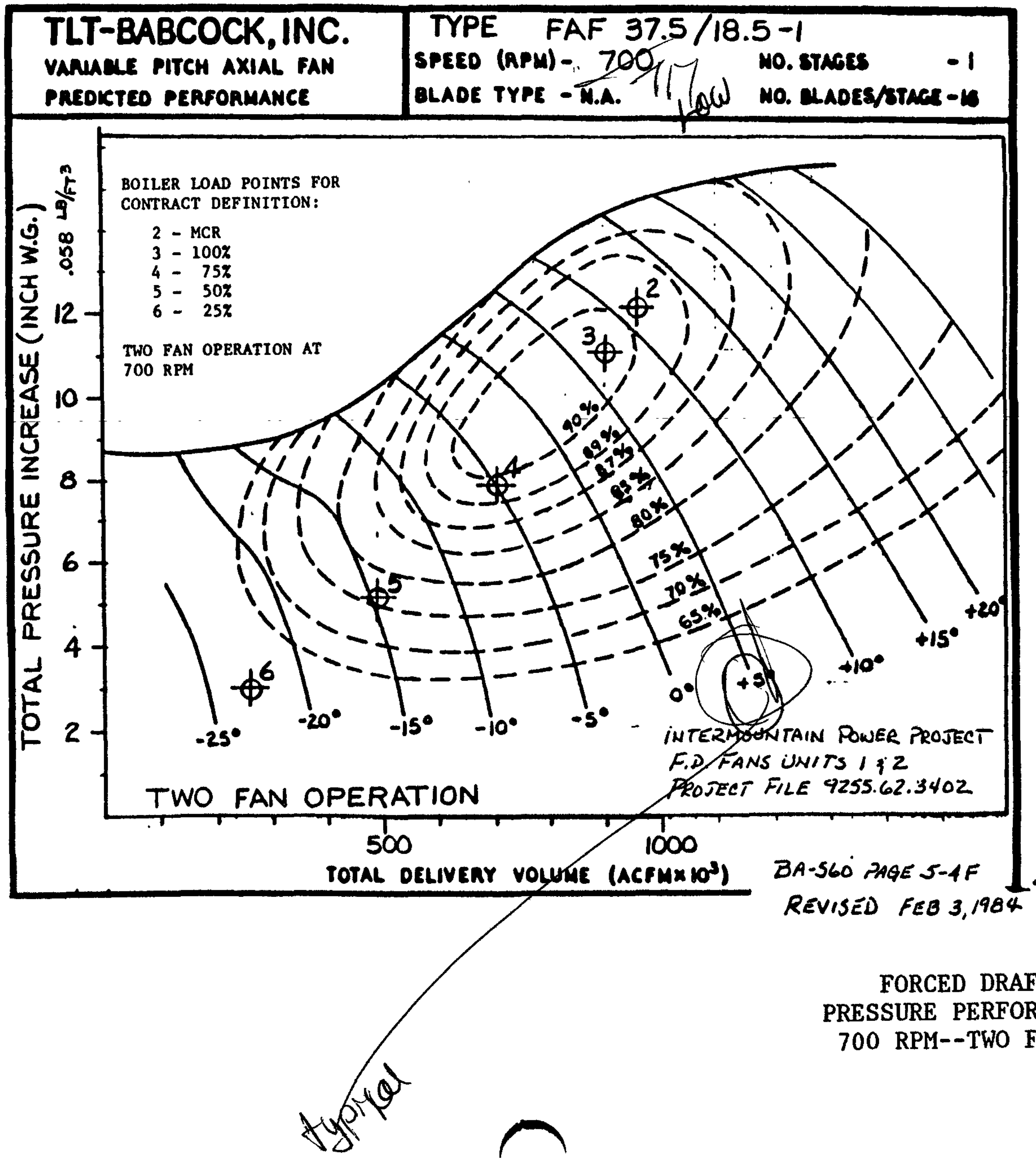


stall around 12" disch press
(have to open something down stream)
abnormal condition

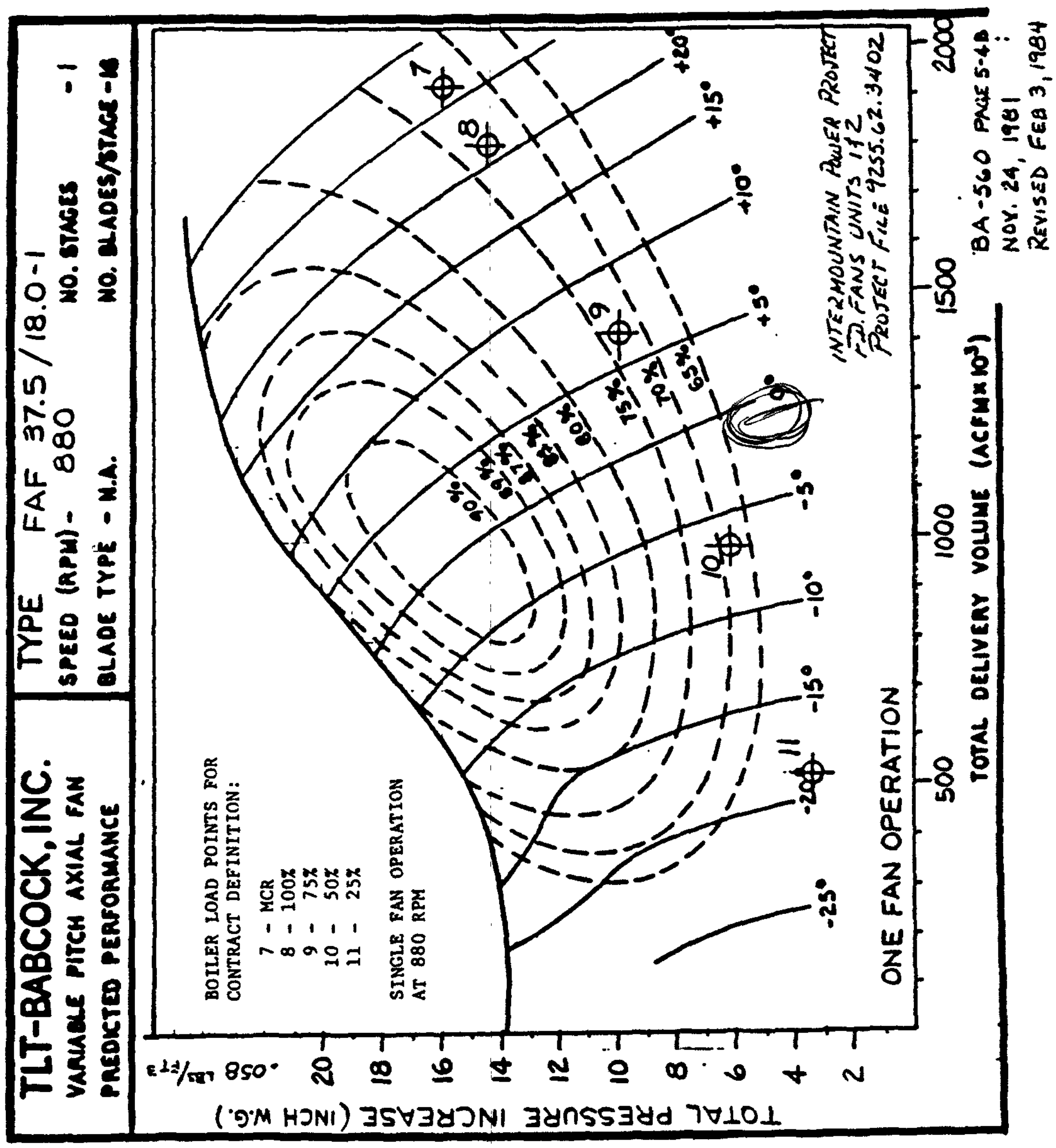
Concern: stall conditions

stroking blades - follower bearing

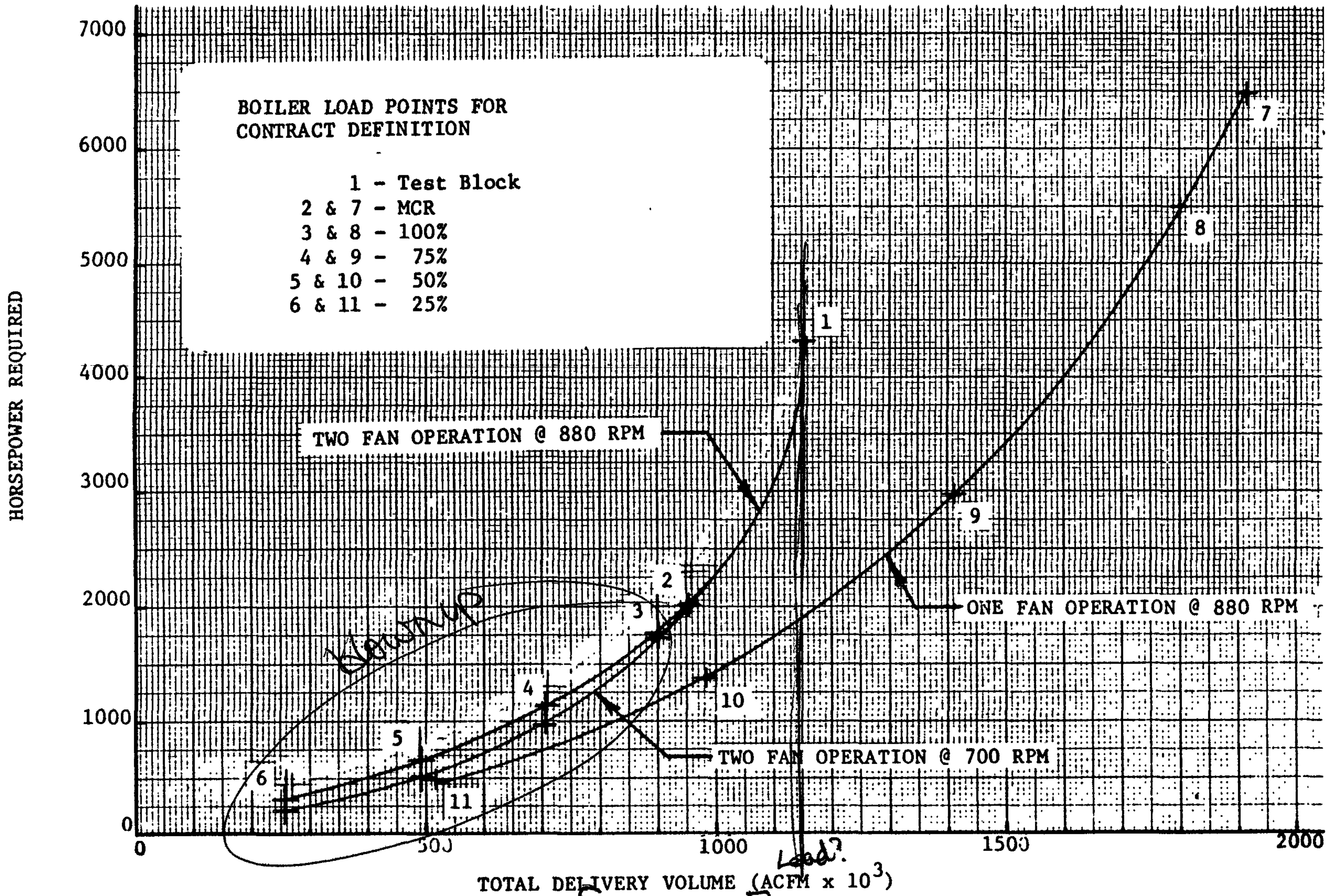
FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
880 RPM--TWO FAN OPERATION
FIGURE 3-1



FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
700 RPM--TWO FAN OPERATION
FIGURE 3-2



FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
880 RPM--SINGLE FAN OPERATION
FIGURE 3-3

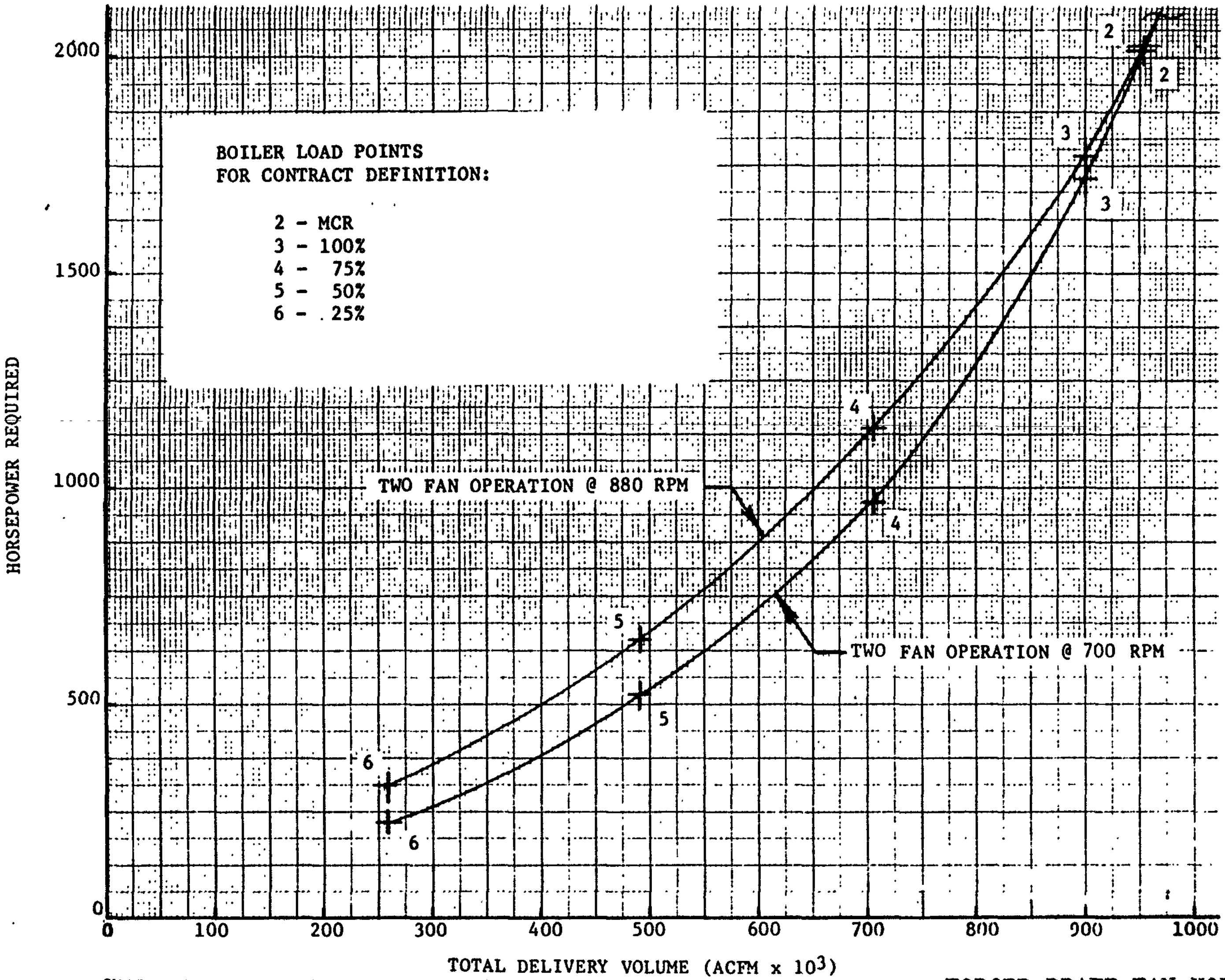


SHOP DRAWING NO. 9255.3402.06-10004 Convert to Load

FORCED DRAFT FAN HORSEPOWER
PERFORMANCE CURVE
FIGURE 3-4

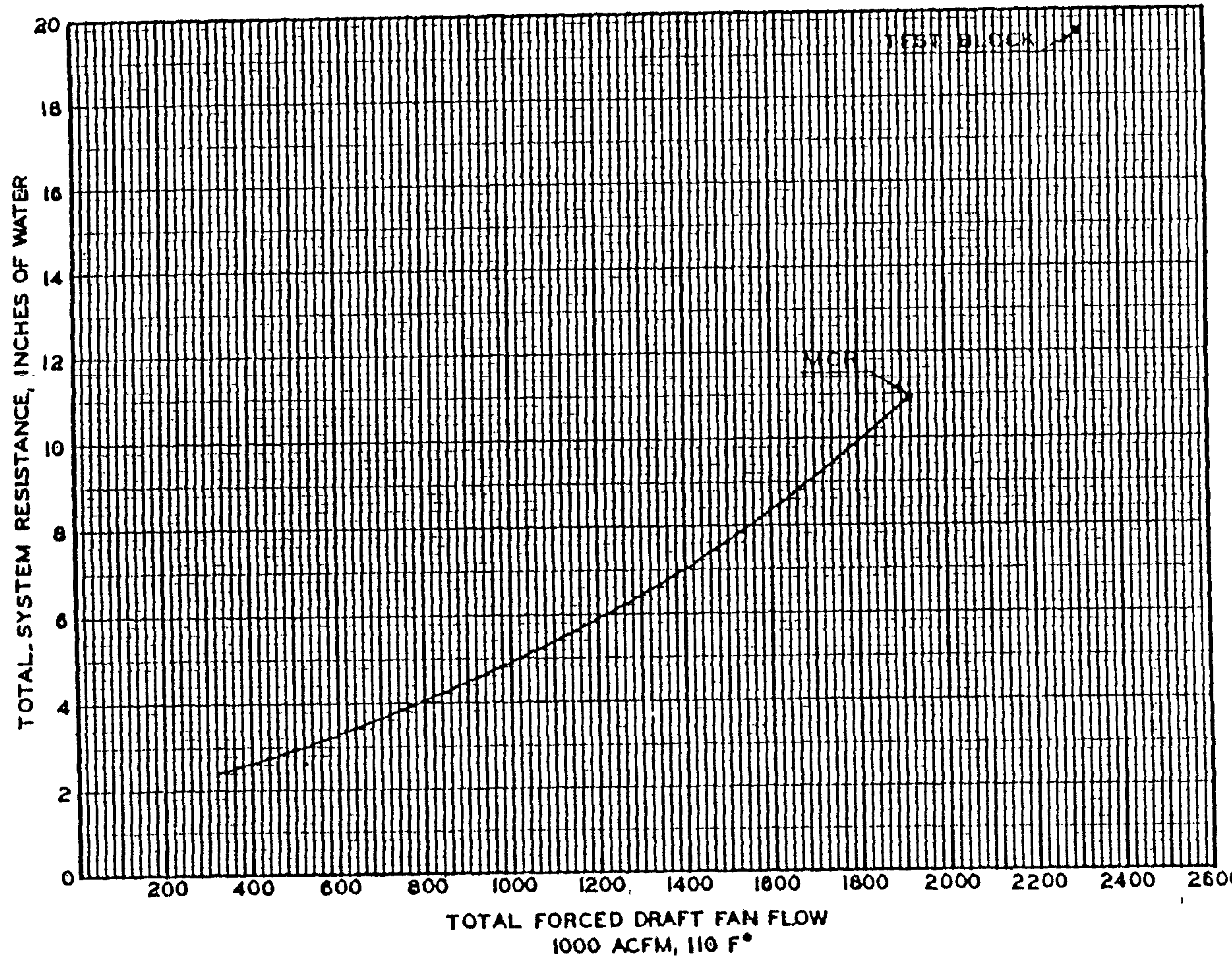
HP vs flow

Use as fan cut of curves



SHOP DRAWING NO. 9255.3402.06-10006

FORCED DRAFT FAN HORSEPOWER
PERFORMANCE CURVE--
OPERATING LOAD POINTS
FIGURE 3-5



NO.	DATE	REVISION	DWN	CK	ACC	APP

BLACK & VEATCH
CONSULTING ENGINEERS
PROJECT
9255

IPP INTERMOUNTAIN POWER PROJECT

FORCED DRAFT FANS SYSTEM CURVE

DM-0032

IP7_038954

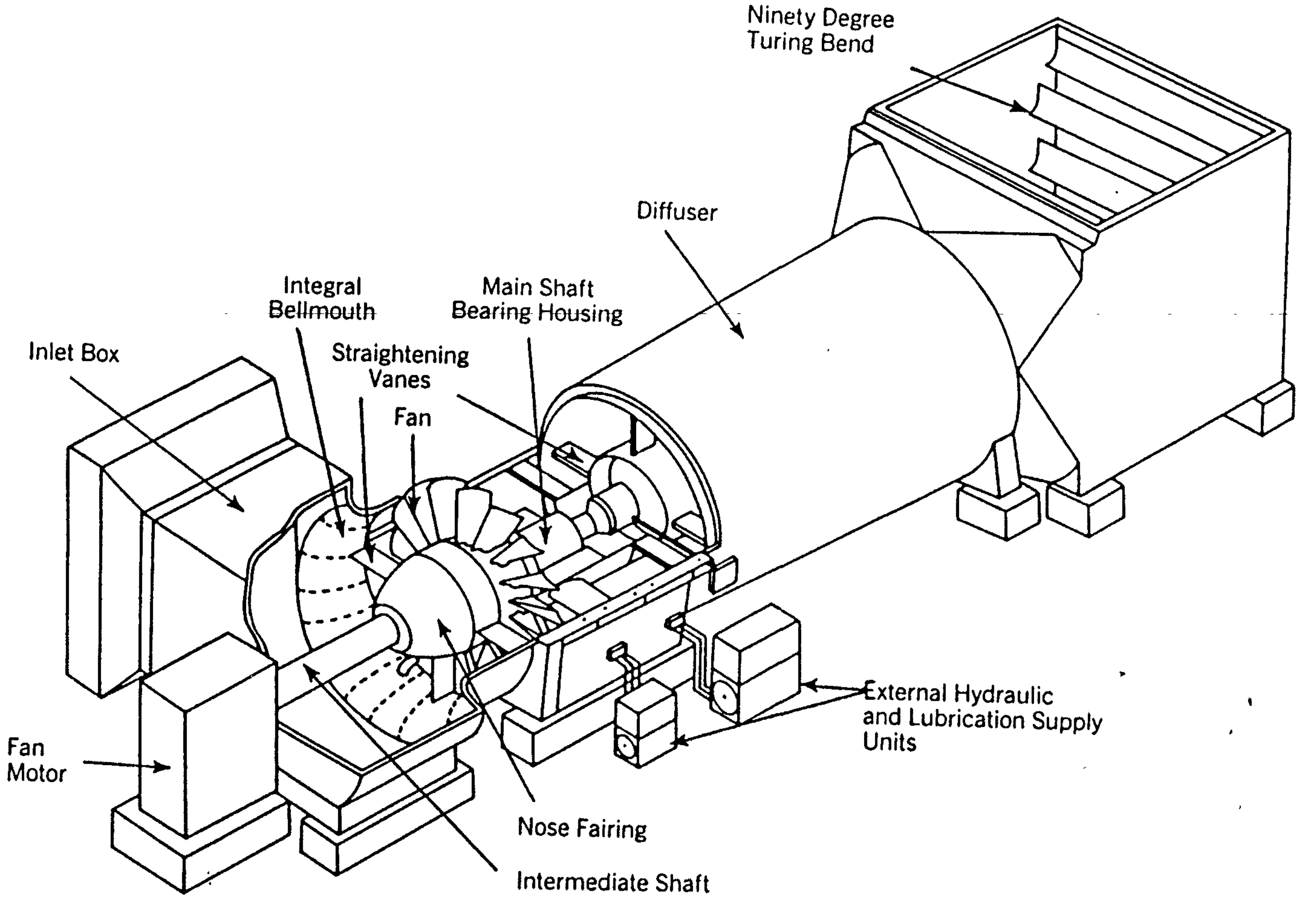


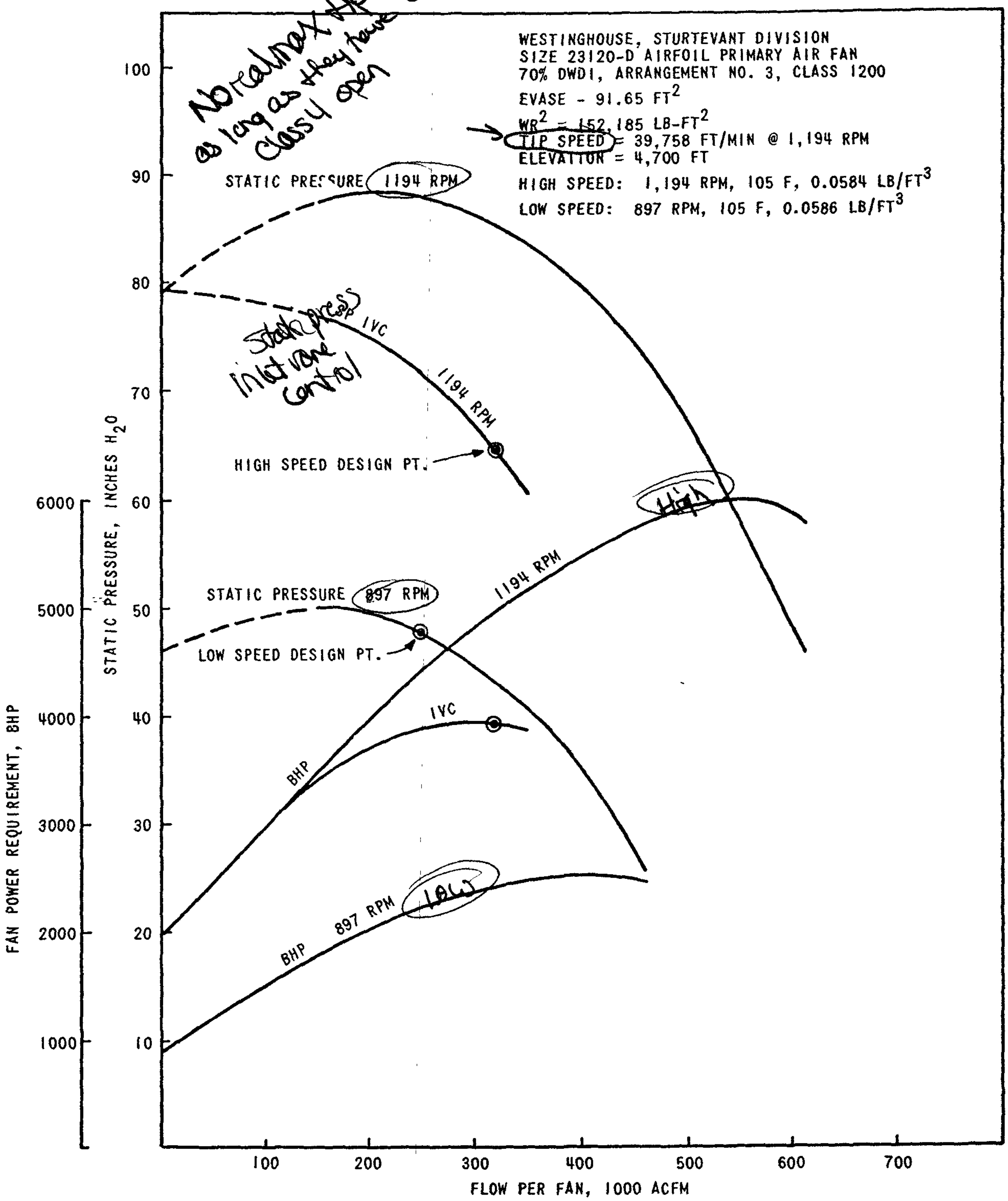
Figure 1

TLT AXIAL FAN
FORCED DRAFT (2)

IP7_038955

TABLE 3-2. PRIMARY AIR FAN PREDICTED PERFORMANCE
Westinghouse-Sturtevant Centrifugal airfoil [2]
2 speed (1194/897 rpm) (3810/2061 HP)

Item	Test Block	MCR
Inlet Air Temperature, F	105	105
Inlet Air Density, lb/ft^3	0.0588	0.0588
Capacity, each fan		
Pounds per hour	1,120,300	882,000
Actual cfm	317,500	250,000
Fan Static Pressure, in. wg	62.5	44.5
Fan Static Efficiency, per cent	81.9	84.9
Design Fan Speed, rpm	1,194	897
Input Horsepower	3,810	2,061



PRIMARY AIR FAN STATIC
PRESSURE AND HORSEPOWER
PERFORMANCE CURVES
FIGURE 3-6

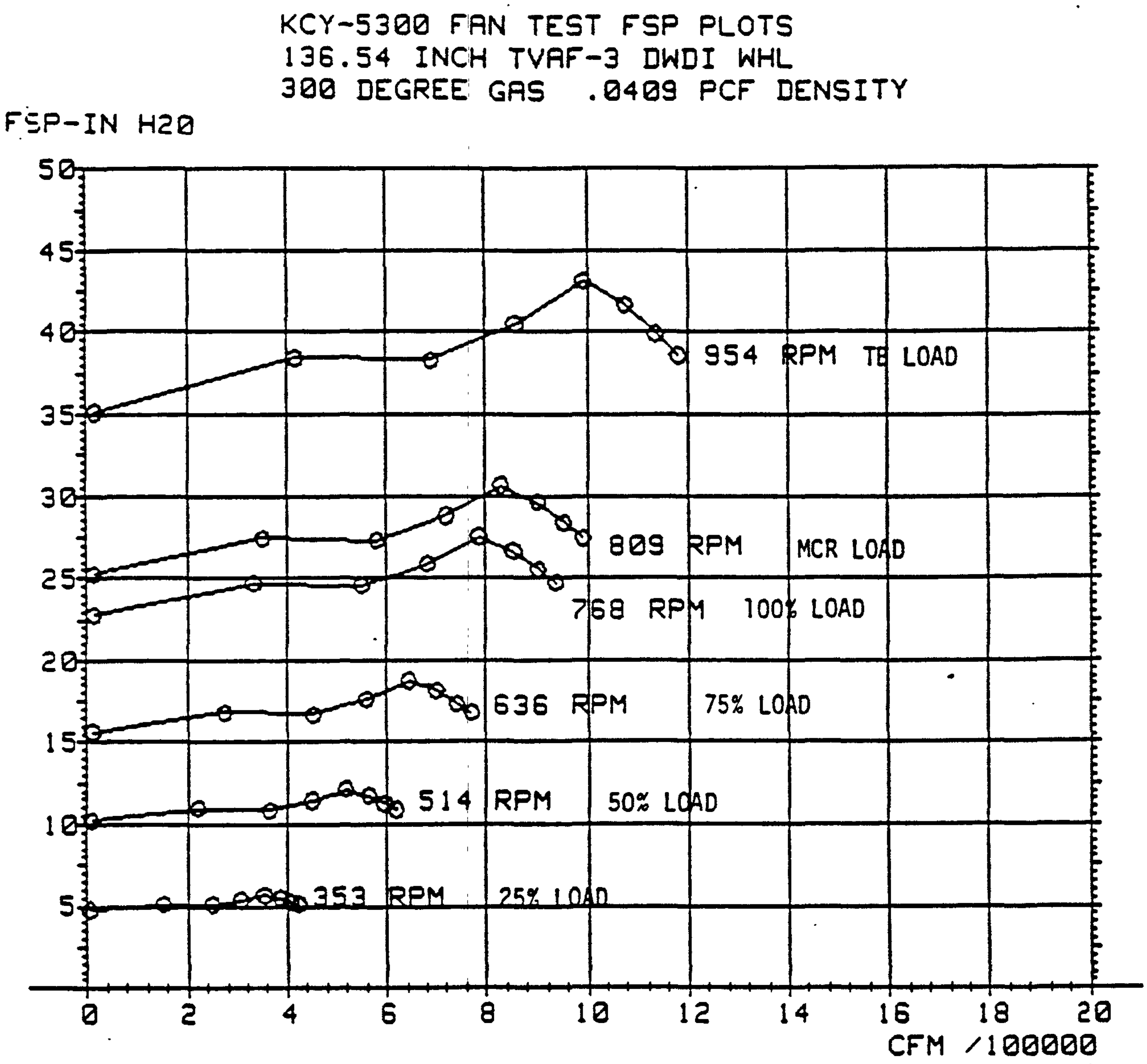
2-speed

ID fans - centrifugal (airfoil, double width, double inlet) 4 ~ 25%
adjustable speed brushless synchronous motor - 7415 HP (W)

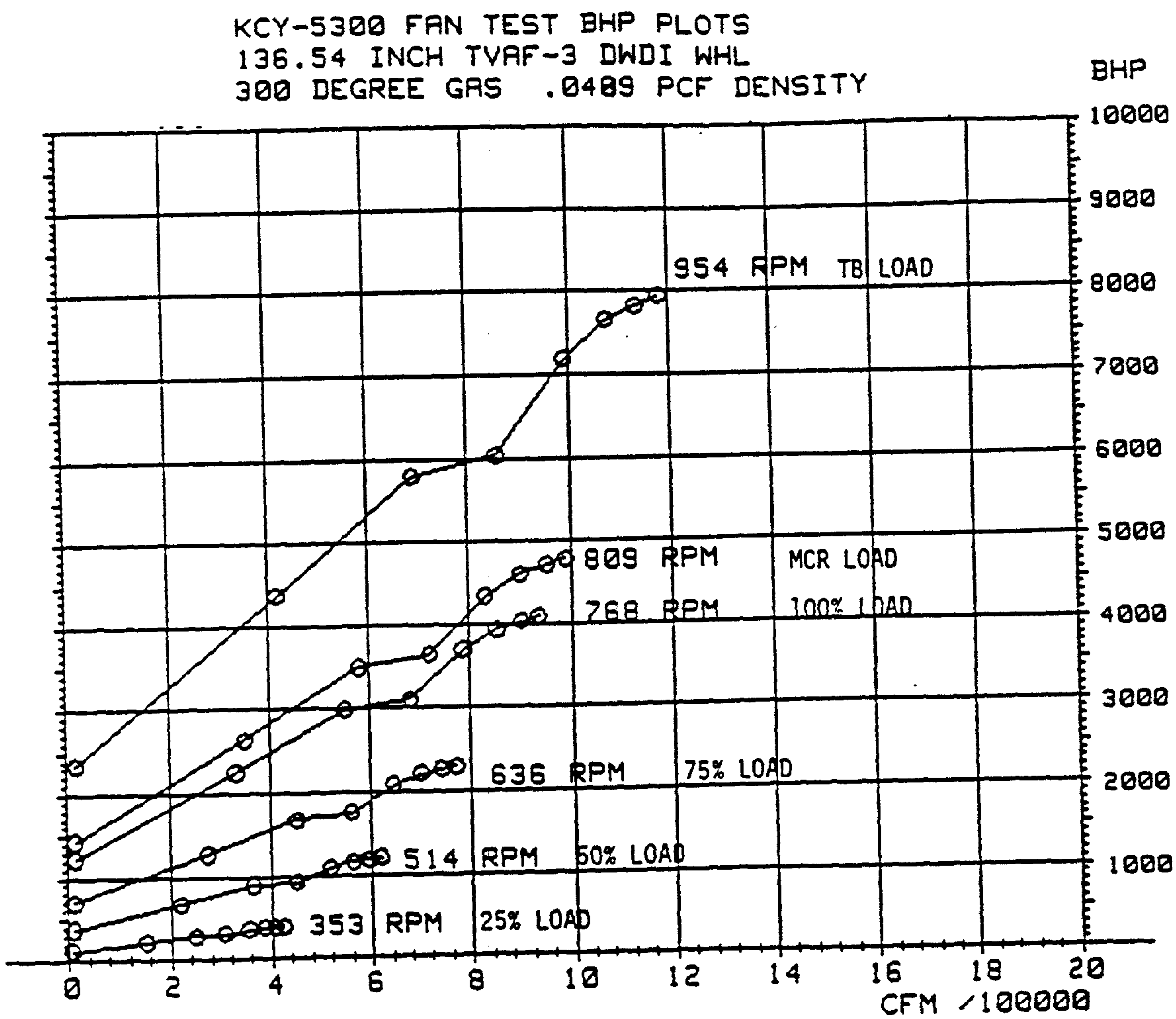
TABLE 3-1. INDUCED DRAFT FAN DESIGN CONDITIONS

Westinghouse - Sturtevant

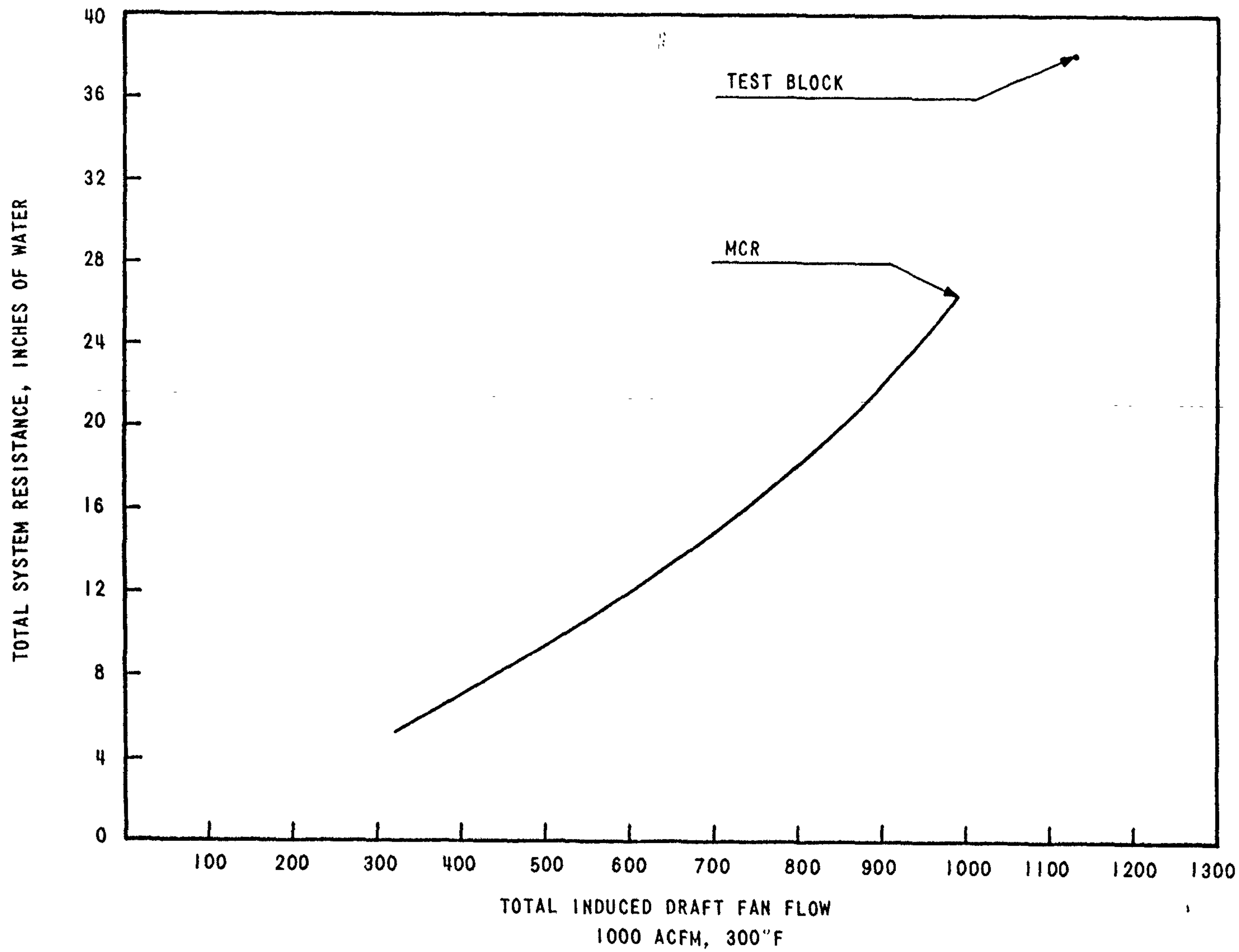
	Generating Unit Load Point					
Item	Test Block	MCR 889 MW	100 820 MW	75 615 MW	50 410 MW	25 205 MW
Inlet Air Temperature, F	300	300	300	300	300	300
Inlet Air Density, lb/ft^3	0.0409	0.0409	0.0409	0.0409	0.0409	0.0409
Capacity, each fan						
Pounds Per Hour	2,769,100	2,436,000	2,291,000	1,852,000	1,363,000	797,000
Actual cfm	1,128,400	992,700	933,600	754,700	555,400	324,800
Fan Static Pressure, in. wg	38.0	26.3	23.8	16.4	11.1	5.3
Fan Static Efficiency, percent	92.18	90.66	90.50	89.89	88.61	83.88
Fan Speed, rpm	954	809	768	636	514	353
Input Horsepower	7,415	4,596	3,918	2,195	1,106	325



INDUCED DRAFT FAN
PERFORMANCE CURVE
FLOW VS STATIC
PRESSURE
FIGURE 3-1



INDUCED DRAFT FAN
PERFORMANCE CURVE
FLOW VS BHP
FIGURE 3-2



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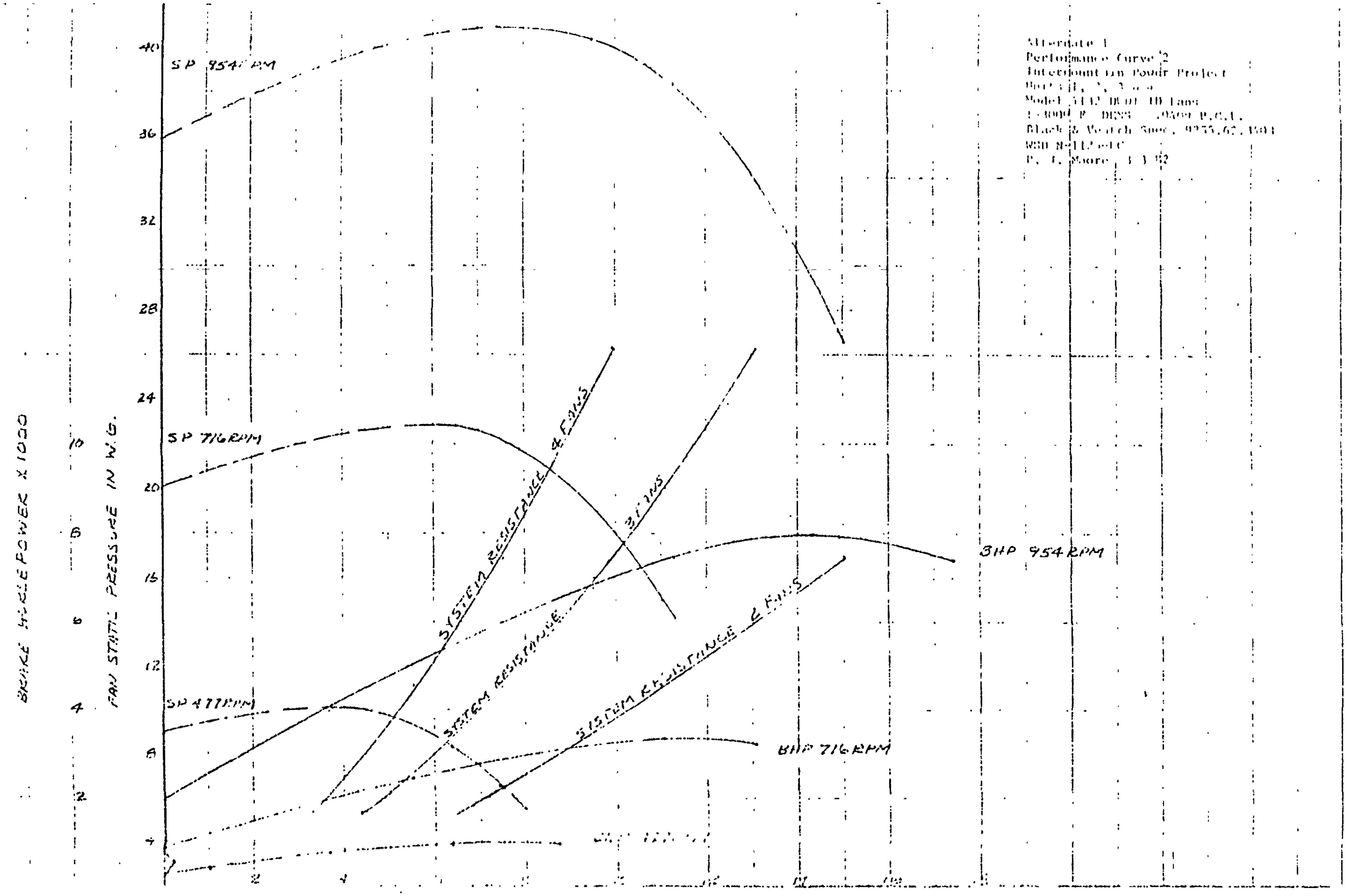
PROJECT
9255

IPP INTERMOUNTAIN POWER PROJECT

INDUCED DRAFT FAN SYSTEM CURVE

DM-0041

IP7_038961


Alternate 1
Performance Curve 2
Intermountain Power Project
Model … Fans
Black & Veatch …
P. …, Moore
SP 954RPM
SP 716RPM
SP 477RPM
BHP 954RPM
BHP 716RPM
SYSTEM RESISTANCE 4 FANS
SYSTEM RESISTANCE 3 FANS
SYSTEM RESISTANCE 2 FANS
FAN STATIC PRESSURE IN W.G.
BRAKE HORSEPOWER X 1000
40
36
32
28
24
20
16
12
8
4
10
8
6
4
2

IP7_038962

FAN NOISE

INTRODUCTION

This paper covers the characteristic noise spectra of a number of typical fan designs and some guidelines for the attenuation of fan noise. The purpose is to furnish typical fan noise values for estimating the noise a fan will radiate to a system and to select attenuation for fan systems. The types of fans covered by this report are centrifugal and axial fans which are normally used with duct systems in central station air conditioning systems, industrial ventilating systems and industrial process applications.

GENERAL DISCUSSION OF FAN SOUND

The proper selection and installation of fans are vitally important. Some of the factors which should be considered are:

1. The air distribution system should be designed for minimum resistance since fan sound generation, regardless of fan type, increases with total pressure.

2. The specific sound power levels of the available fan designs for any given job should be examined by the design engineer. Different types of fans generate different levels of sound and produce different octave band spectra. The engineer should select a fan which will generate the lowest possible sound level. The selection of parameters for a particular fan installation includes many other factors than noise, and in most cases, these other factors will narrow down the types of fans available for a given job. However, the engineer should examine those available types in order to select the lowest noise level.

3. The fan should be selected to operate near its maximum efficiency point when handling the required air quantity and total pressure. Proper sizing of the fan is important in assuring a minimum of fan sound for any given type of fan. Sound generation and efficiency are not directly related on an energy basis because of the exceedingly small amount of energy required to produce sound. However, the same factors which reduce fan efficiency also increase sound.

   Oversizing fans increases the sound generated by the fan. The increase in sound power level results from a decrease in fan efficiency because the fan is too large.

   An undersized fan will also be lower in efficiency and sound power levels will be higher than for the optimum size fan.

4. Duct connections at both the fan inlet and outlet should be designed for uniform and straight air flow. Swirling inlet air flow particularly should be avoided. Variations from accepted application arrangements can severely degrade both the aerodynamic and acoustic

performance of any fan type and invalidate manufacturers' rating or other performance predictions.

FAN NOISE TRANSMISSION

Previous literature on noise control discussed a very useful concept for the solution of noise control problems by stating, "The key to practical noise control is that every acoustic problem can be broken down into three types of functional components". Ref. 1.

SOURCE -------> PATH -------> RECEIVER

While a breakdown into these three components may seem to be an over-simplified expression of the actual problem, it is in reality a good description of the problem and an excellent model for working on fan noise control problems.

This simplifies schematic illustrates the scope of the discussion in this paper. As fan manufacturers we can present data on the noise produced by the source (fan) but cannot establish the acoustical criteria for the receiver.

The path may be divided responsibility.

In actual problems there will be more than one path between the source and the receiver. These paths may be in series or in parallel.

SOURCE----->PATH------> PATH--------> RECEIVER

In the case of fan noise control, the diagram would appear as:

FAN-------> PATH--------> PATH--------> RECEIVER

In most cases the paths between the fan and receiver consist of such factors as ductwork, walls, floors, room treatment and distance to property lines. The fan manufacturer has no control over these paths. However, in those cases where attenuation equipment is a part of the fan package, a portion of the path becomes the responsibility of the fan manufacturer.

FAN--------> ATTENUATOR----------> PATH----------> RECEIVER

Fan noise measurement techniques are specified by the Air Movement and Control Association (AMCA Standard 300-85) Test Code for Sound Rating Air Moving Devices.

This test code calls for fan noise to be reported in terms of sound power levels in eight octave bands and for the laboratory noise measurements to be taken in a reverberant room using a calibrated reference sound source and the substitution technique. The fan industry reports fan noise in terms of sound power levels in 8 octave bands (63, 125, 250, 500, 1000, 2000, 4000, 8000 Hz).

At the present time AMCA Standard 300-85 is intended to apply to the following types of fan equipment (1) Central Station air conditioning and heating and ventilating units, (2) Centrifugal fans, (3) Industrial,axial, and propeller fans, and (4) Power roof and wall ventilators.

SYMBOLS

L_W	= Sound Power Level, dB
K_W	= Specific Sound Power Level, dB
L_P	= Sound Pressure Level, dB
RPM	= Revolutions per minute
B_f	= Blade frequency
BFI	= Blade Frequency Increment
Q	= Flow Rate, cubic feet per minute
P	= Pressure, inches of water gage

SUBSCRIPTS

W	= Power level
P	= Pressure level

SOUND POWER LEVEL

It is essential to understand the difference between sound power level and sound pressure level.

Figure 1 shows a typical situation for many conventional ventilating systems. In this case, the sound power level generated by the fan results in a sound pressure level in the fan room that may be objectionable, depending on whether or not the fan room is normally occupied. In shop the space, the fan noise is not objectionable since the intervening wall provides sufficient attenuation and there is no opening in the ductwork. Also, the noise level requirement in this shop area is not severe. In the office farther from the equipment room, the intervening walls have provided sufficient attenuation, but the fan noise is transmitted through the ductwork and radiated from the discharge grille. This occupant finds the sound pressure level too high for good working conditions.

The fan noise is also transmitted through the fresh air inlet louvers to persons outside the building. A person close to the louvers may find the sound pressure objectionable, but since noise levels decay rapidly with distance, persons farther away will not be aware of the fan.

This illustrates why the question "Will this fan meet acceptable noise criteria?" cannot be answered by the fan manufacturer. The answer depends on where the listener is located. Fan noise from the same source will be described in various ways by various listeners, since human ears respond only to the sound pressure in their particular environment. The essential factor to observe is that the sound power radiated by the fan is constant, but the resulting sound pressure is a function of the acoustical environment.

The fan manufacturer can give accurate data on the sound power generated by the fan. It is then up to the system designer or the acoustical consultant to calculate the sound pressure level that will result in the particular environment of the job under consideration.

FAN DESIGN AND NOISE CHARACTERISTICS

The primary purpose of a fan in any air-moving system is to move a given quantity of air against a given pressure differential as efficiently as possible, and it must do this at a reasonable first cost. Some fans will have secondary requirements, such as ability to handle dust-laden air, resistance to abrasion, a construction suitable for production techniques, or a construction that can be repaired easily in the field. Only after these requirements have been satisfied is the fan evaluated from a noise standpoint.

All discussions of fan noise must take into account this relative importance of noise in the overall fan design problem.

It must be understood that every fan generates an amount of noise that is proportional to the volume flow rate, the pressure developed, and the type of fan. The system design engineer must accept this noise as a part of his engineering design problem. It is impossible to design a large, high speed, high pressure fan that generates only low sound levels. Fan noise is just as much an integral part of the fan performance as is the horsepower requirement. One cannot arbitrarily establish the horsepower requirements for a given fan since the horsepower is determined by the actual operating requirements. Fan noise is a function of these same requirements and noise levels cannot be set at arbitrary values but must be based on actual operating requirements.

No single type of fan will solve all fan problems. If this were the case, obviously, only that type would be offered by fan manufacturers. However, many different types are required to satisfy the many different fan applications and, for a truly adequate engineering analysis of each system, all applicable fan types should be considered. For example, the radial fan shown in Table I with straight radial blades is probably one of the noisiest fans in common use, since it has not only a high noise level, but also a predominant blade frequency tone that can be extremely objectionable. Consideration of noise alone would eliminate this fan from use, but it is a very common industrial fan. It is used in cases where erosive material is to be handled, it is a rugged design, it does not collect dirt on the blades, and it can be easily repaired, since only flat plates are required.

Similar reasons exist for the widespread use of other noisy fan designs. Fans are seldom, if ever, designed exclusively on an acoustical basis. The fan is designed for the required duty, and if it is the best fan for that duty, the fan noise will be an integral part of that design and will be the minimum for that specific application. There is no magic to fan noise, and minor alterations to a good fan design will have no effect on the noise.

There are no abrupt changes in the noise characteristics of fans as the design of the fan is altered slightly to produce a small change in fan performance. The fan noise characteristics also change gradually. For example, it has been explained that the radial blades fan is probably one of the worst fans from a noise standpoint. By a change in blade shape, such as the redial tip-forward curved heel design, the fan noise and especially the blade frequency components are lowered. However, it is emphasized that his is a matter of gradual improvement and the noise level does not drop abruptly, and although the blade frequency component is lowered, it is <u>not</u> eliminated.

FAN NOISE DATA

The information in Table I is given on the principal types of fans used in commercial and industrial installations. The fan noise is listed in terms of specific sound power levels.

Specific Sound Power level is defined as the sound power level generated by a fan operating at a flow rate of 1 CFM, and a pressure of 1 inch of water. By reducing all fan noise data to this common base, the specific sound power level concept allows direct comparison of the octave band levels of various fans and serves as a basis for a convenient method of estimating the noise levels of fans at actual operating conditions.

The Specific Sound Power levels shown in Table 1 represent the noise generated by the fan when it is operating at an efficient point on the performance curve. The data represent the results of tests on fans obtained from a number of sources, and are representative of commercially available fans which follow the principles of good design.

Fans generate a tone at blade frequency and the strength of this tone depends, in part, on the type of fan. In order to account for this blade frequency, an increase should be made in the octave band into which the blade frequency falls. The amount of increase to be added to this band is listed on Table I for each fan as "Blade Frequency Increment" (BFI).

$$BF = \frac{\text{NO. OF BLADES} * \text{RPM}}{60}$$

The number of blades and the fan RPM can be obtained from the catalog being used for fan selection.

ESTIMATING FAN NOISE

For preliminary engineering work it is not necessary to obtain the actual sound power levels from the fan manufacturer since reasonable accuracy can be achieved by use of the information found in the SPECIFIC SOUND POWER LEVEL table .

This method of fan noise estimation is based on the assumption that well-designed fans which have been properly selected for size and speed are being used and, therefore, the fans are generating the minimum noise level that can be seasonably expected under the required operating conditions. Obviously, poorly designed fans will make more noise under these same conditions.

The design engineer must realize that further reduction of fan noise can be accomplished only by adding attenuation to the path between the fan and the receiver.

The Specific Sound Power levels listed in Table I provide the basis for a method for estimating the sound power levels of fans under actual operating conditions. The principal correction is a function of the flow rate (Q) and the pressure rise across the fan (P).

$$\text{CORRECTION} = 10\ \text{LOG}\ (Q) + 20\ \text{LOG}\ (P)$$

To estimate the operating power levels it is necessary to determine:

1. Fan Type
2. Q - Flow Rate - cubic feet per minute
3. P - Total Pressure - inches of water
4. Fan speed - RPM
5. Number of blades in wheel

By obtaining the specific sound power level from Table I for the type of fan selected and applying the correction factor for the operating CFM and pressure, and applying the blade frequency increment, the sound power level at operating conditions may be estimated as in the following example:

<u>EXAMPLE 1</u>

Estimate the sound power level radiated from the inlet of a backward curved blade fan operating at 50,000 CFM and 3.1" H_2O. The wheel diameter is 44.5".

Step 1. The specific sound power level of the backward curved blade type fan is determined from Table I.

Step 2. Using Equation 3 calculate the correction factor to be added for 50,000 CFM and 3.1" H_2O.

(10 log CFM + 20 log P)
(10 log 50,000 + 20 log 3.1)
(47 + 10)
57 dB

Step 3. Catalog information tells us this fan has 16 blades and will operate at 500 RPM to meet the operating requirements. Using Equation 2:

$$BF = \frac{16 * 500}{60}$$

133 Hz falls in the 125 Hz octave band

Table I lists a blade frequency increment of 3 dB for the backward curved blade fan so this value should be added to the 125 Hz octave band.

Step 4. Values from Steps 1 through 3 are total sound power levels. To calculate the sound power level from the inlet (or outlet) alone, subtract 3 dB from the above values.

Step 5. Combining these steps:

Octave band	63	125	250	500	1000	2000	4000	8000
From Step 1	35	35	34	32	31	26	18	15
From Step 2	57	57	57	57	57	57	57	57
From Step 3		3						
From Step 4	-3	-3	-3	-3	-3	-3	-3	-3
						Estimate		
Sound Power (One Side)	89	92	88	86	85	80	72	69

FAN NOISE SPECIFICATIONS

As mentioned above, the noise of fans is recorded in terms of sound power levels and an understanding of the sound power level concept is essential to the understanding and engineering of air-handling systems. Although the fan noise is reported in terms of sound power levels, the noise criteria for system is expressed in terms of sound pressure levels. Since the fan manufacturer has no knowledge of the type of acoustical environment in which the fan will be located, it is impossible for the fan manufacturer to predict what the sound pressure level will be under operating conditions of his fans in the field.

The design engineer, or acoustical consultant, must use the sound power levels as furnished by the fan manufacturer, and, applying the principles found elsewhere in this manual, must calculate the resulting sound pressure level in the particular environment of interest.

The National Institute of Safety and Health (NIOSH) establishes maximum limits of exposure to noise for employees of industrial and commercial establishments. It is essential to recognize that these requirements apply to the acoustical environment in which people are working and do not establish maximum noise levels for fans. It is incorrect to specify the noise requirements of a fan by stating that "The sound pressure of the unit shall comply with the NIOSH requirements of 90 dBA." Obviously, the fan manufacturer cannot relate to this requirement because he does not know where the employees will be located, not what the acoustical environment of the fan and employees will be.

NIOSH compliance is determined by measuring sound pressure levels at the employees' work station and not by measurements at some arbitrary distance from the fan.

SPECIFICATION FORMAT

Design engineers may write specifications covering the noise generated by a fan in either of two general forms:

1. Specification Form No. 1

   A request for the sound power level generated by the fan when it is operating at the specified conditions.

2. Specification Form No. 2

   A specified upper limit of the sound power level that will be permitted to be generated by the fan under the operating conditions.

SPECIFICATION FORM NUMBER 1

In this form the design engineer is requesting the total sound power level generated by the fan which will permit him to calculate the resulting sound pressure level in his particular fan installation. If these pressure levels are too high for his installation, he must select appropriate sound attenuation facilities to be added to the system. A specification aimed at this type of requirement may be written as follows:

> "NOISE GENERATED BY THE FAN WHEN OPERATING AT THE SPECIFIED VOLUME FLOW RATE AND PRESSURE SHALL BE DETERMINED ACCORDING TO THE CONDITIONS OF AMCA STANDARD 300-67, TEST CODE FOR SOUND RATING, AND SHALL BE REPORTED IN TERMS OF SOUND POWER LEVEL RE 10^{-12} WATT IN EIGHT OCTAVE BANDS."

SPECIFICATION FORM NUMBER 2

The second method of specification is used in those cases where the design engineer has calculated the acoustical properties of his system and has determined the maximum sound power level that can be permitted on this particular job. The design engineer then specifies the maximum sound power limits for the fan and it is up to the fan manufacturer to provide the necessary attenuation on his fan to meet these levels. In this case the fan and the attenuation facilities are considered to be one package. Obviously, the addition of the attenuation facilities adds to the cost of the fan.

Specifications based on this approach may be written as follows:

> "NOISE GENERATED BY THE FAN WHEN OPERATING AT THE SPECIFIED VOLUME FLOW RATE AND PRESSURE SHALL BE DETERMINED ACCORDING TO THE CONDITIONS OF AMCA STANDARD 300-67, <u>TEST CODE FOR SOUND RATING</u>, AND SHALL BE REPORTED IN TERMS OF SOUND POWER LEVEL RE 10^{-12} WATT IN EIGHT OCTAVE BANDS AND SHALL NOT EXCEED THE LIMITS SHOWN IN THE TABLE II."

TABLE II

Octave Band Center Frequency	Hz	63	125	250	500	1000	2000	4000	8000
Maximum sound Power levels	x	x	x	x	x	x	x	x	x

(Where the x's appear in the above table the design engineer simply inserts the maximum sound power levels permissible on his particular job.)

<u>COMMON FAULTS IN PRESENT SPECIFICATIONS</u>

In some cases, noise specifications have been written to require sound pressure level reading at specifies points around the fan. In most instances, this has resulted in completely unsatisfactory reading for any engineering requirements. Readings taken close to the fan are in the near field and are unreliable especially in the low frequency bands. Great variation in the reading can be obtained by moving the microphone a relatively short distance. It does no good to specify that the highest readings are to be recorded since these readings are no more significant than any other readings. In addition, this technique ignores the sound energy that is radiated to the duct system which, in many cases, is the most important noise path in a ventilation system.

The Occupational Safety and Health Act (OSHA) establishes maximum limits of exposure to noise for employees of industrial and commercial establishments. It is essential to recognize that the OSHA requirements apply to the acoustical environment in which <u>people</u> are working and do not establish maximum noise levels for fans. It is incorrect to specify the noise requirements of a fan by stating that "the sound pressure of the unit shall comply with the OSHA requirements of 90 dBA".

OSHA compliance is determined by measuring sound pressure levels at the employees' work station and not by measurements at some arbitrary distance from the fan.

Fan noise attenuation can be expensive, especially if unreasonable levels are specified. Unless such levels are absolutely essential, it is a waste of money to specify low levels "just to be safe".

To specify low sound pressure levels in the vicinity of the fan makes very little sense when, perhaps, a man enters this area once a day for a very short time simply to check the operation of the fan. The noise criteria should be established at the point where the nearest listener will be for significant periods of time. For example, it makes no sense to have a quiet equipment room in the building when there is no regular occupancy in this room. The design should be based on the requirement of the nearest occupied space.

FAN NOISE ATTENUATION

Previous sections of this chapter have outlined a procedure to estimate the sound power level that will be radiated by various types of centrifugal and axial flow fans. These values represent reasonable power levels for these various types of fans and it is unlikely that significantly lower levels will be generated by the basic fan configuration. If lower power levels are required, it is necessary to add this attenuation to the basic fan. This attenuation can be added as separate units in the field, or may be added to the fan as an integral part of the fan assembly.

Although specific details of fan noise attenuation will vary, the principles are demonstrated in Figures 9.3 and 9.4. In Figure 9.3 a centrifugal fan is shown with attenuation on both the inlet and outlet. This fan is illustrated as a supply air fan in a central station ventilating system. The discharge of the fan has been fitted with an attenuator to reduce the amount of noise radiated from the discharge of the fan to the supply air ductwork. This attenuator is sized by using the basic sound power levels of the fan (without attenuator) to calculate the noise level that would result in the room with the most critical criteria. The amount by which the calculated level exceeds the allowable level is the amount of noise that the fan attenuator must remove. Attenuators are rated in decibel reduction and these decibel reduction values can be used to calculate the reduction in sound power level of the fan.

There is no "rule of thumb" for selecting such attenuators and each attenuator must be based on the actual requirements for that particular fan. In this way the noise level radiated by the discharge of the fan can be brought down to any reasonable level.

Figure 9.3 also shows noise attenuation on the fan intake system. In this case it has been predicted that the fan noise radiated by the suction side of the fan would be objectionable outside the fresh air intake. This attenuator is selected in precisely the same way as the one mentioned in connection with the fan discharge. However, in this case outdoor noise criteria must be used and it is quite likely that this level would be established by applying the criteria to either the property line of the building or at the nearest building or residence.

The walls of the fan equipment room have been indicated as heavy construction in order to prevent transmission of the equipment room noise to the adjacent office spaces.

The best way to mount rotating equipment such as a fan and the motor is to bolt it directly to a foundation set in the earth. The next best way is to design the structure so that it is not resonant ith any of the exciting frequencies present in the equipment and bolt the rotating equipment directly to the structure. This requires a vibration analysis of the structure but that is a practical requirement with the computer programs and equipment that are available today.

As an alternative the entire fan assembly may be supported on an inertia base which is isolated from the equipment room floor by vibration isolators. The simple act of putting springs under the equipment is is not an adequate treatment for vibration isolation. The inertia base and springs must be properly engineered and installed to be effective. This type of isolation is not fooproof.

The fan is also equipped with a vibration isolator in the discharge section between the fan and the continuing ductwork to prevent the transmission of vibration from the fan casing to the attached ductwork. Here again, unless the vibration break is used, the vibration from the fan can initiate resonant vibrations in the ductwork which can be quite annoying from an acoustical standpoint.

Figure 9.4 demonstrates the same principles of fan noise attenuation, but for an axial flow fan. In this case the attenuators are attached directly to both the upstream and downstream sides of the fan and the two noise attenuators and the fan and the two noise attenuators and the fan are treated as a unit. This unit is separated from both the inlet and discharge ductwork by vibration breaks and the entire unit is supported on a slab which, in turn, is isolated from the building structure by vibration mounts.

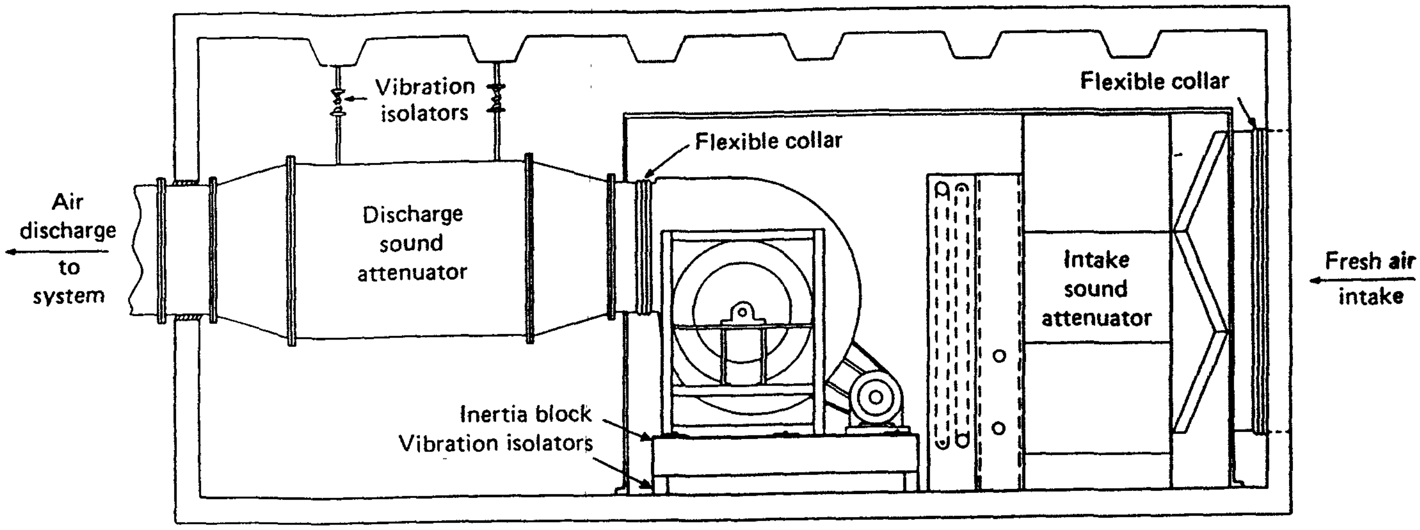

Vibration isolators
Flexible collar
Flexible collar
Air discharge to system
Discharge sound attenuator
Intake sound attenuator
Fresh air intake
Inertia block
Vibration isolators

Fig. 9.3

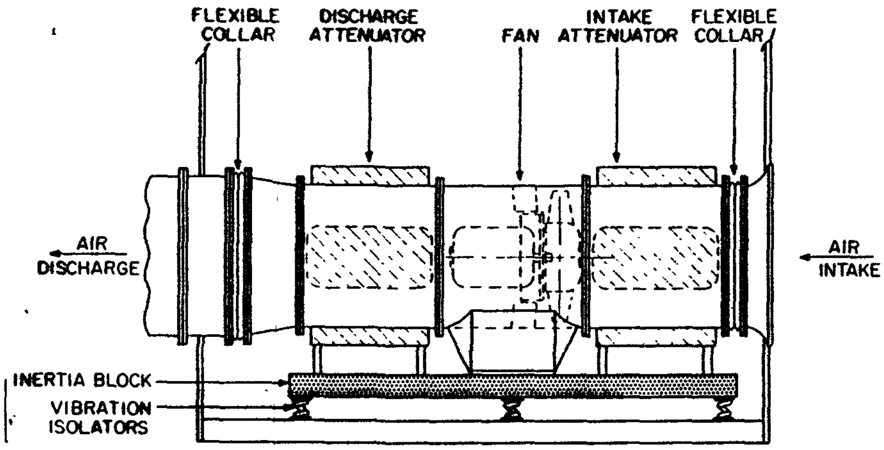

FLEXIBLE COLLAR
DISCHARGE ATTENUATOR
FAN
INTAKE ATTENUATOR
FLEXIBLE COLLAR
AIR DISCHARGE
AIR INTAKE
INERTIA BLOCK
VIBRATION ISOLATORS

Fig. 9.4

$\dot{V}_{REQUIRED}$ AT FAN DIS. = 33,000 ACFM

PERFORMANCE CURVE

nyb | The New York Blower Company

To determine performance at another RPM multiply:

1- CFM x K
2- SP x K^2
3- BHP x K^3

where K is new RPM divided by RPM shown at right.

Customer's No. 36XX-20690-20707-40
Customer FLEXKLEEN CORP.
Tagging SEE BELOW (NARROW WIDTH WHEEL)
Size 40-93.5%NW Type PLR CL IV WHEEL
37950 CFM at 18.5 " SP at 0.0768 D. (DENSITY)
at 1780 RPM at 147.6 BHP
File No. Y-1814S Date 17-APR-83

INTERMOUNTAIN GEN. STATION
B & V PROJECT 9255
B & V FILE 61.0405.05-90956

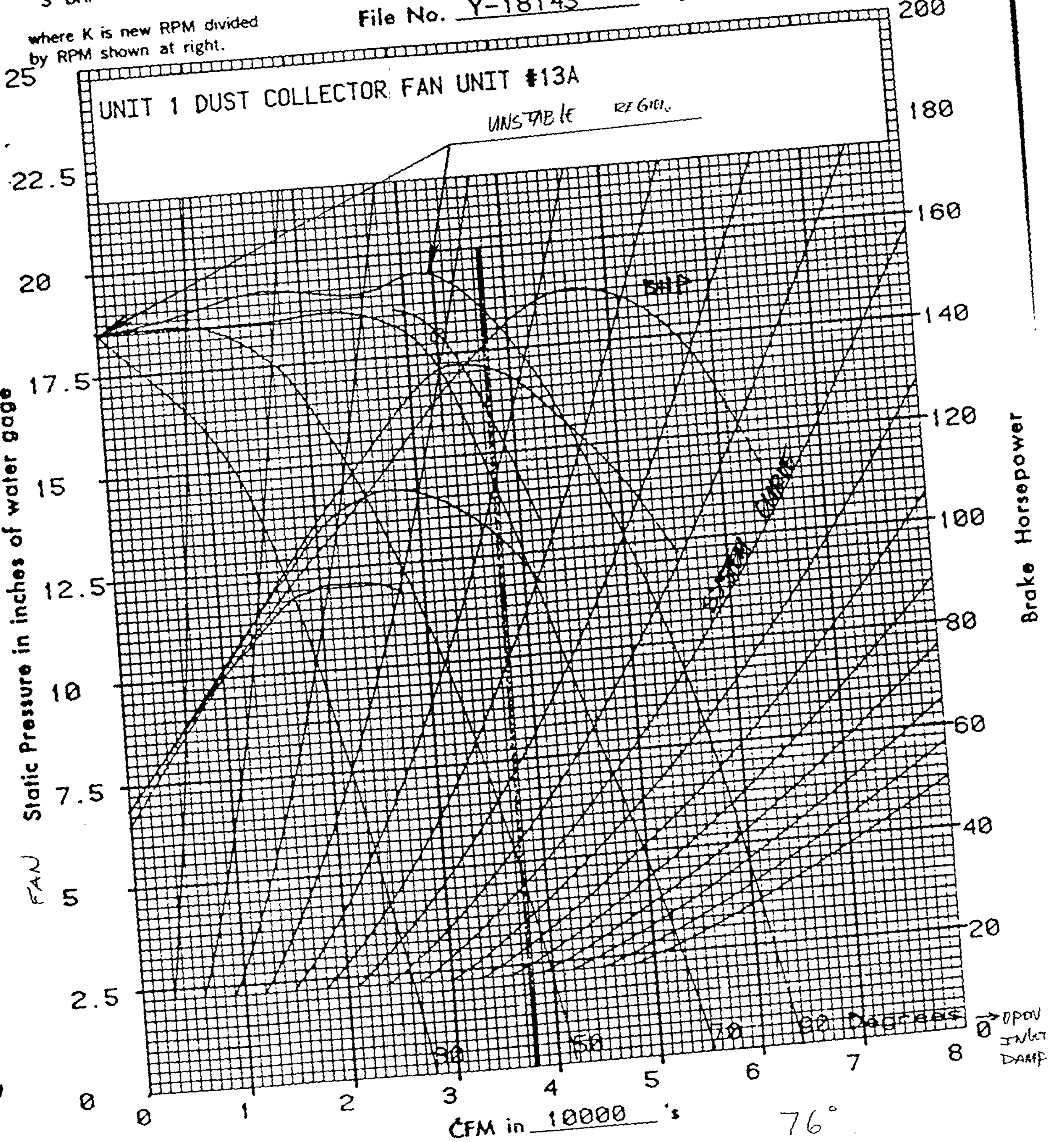


FIG. 11

IP7_038975

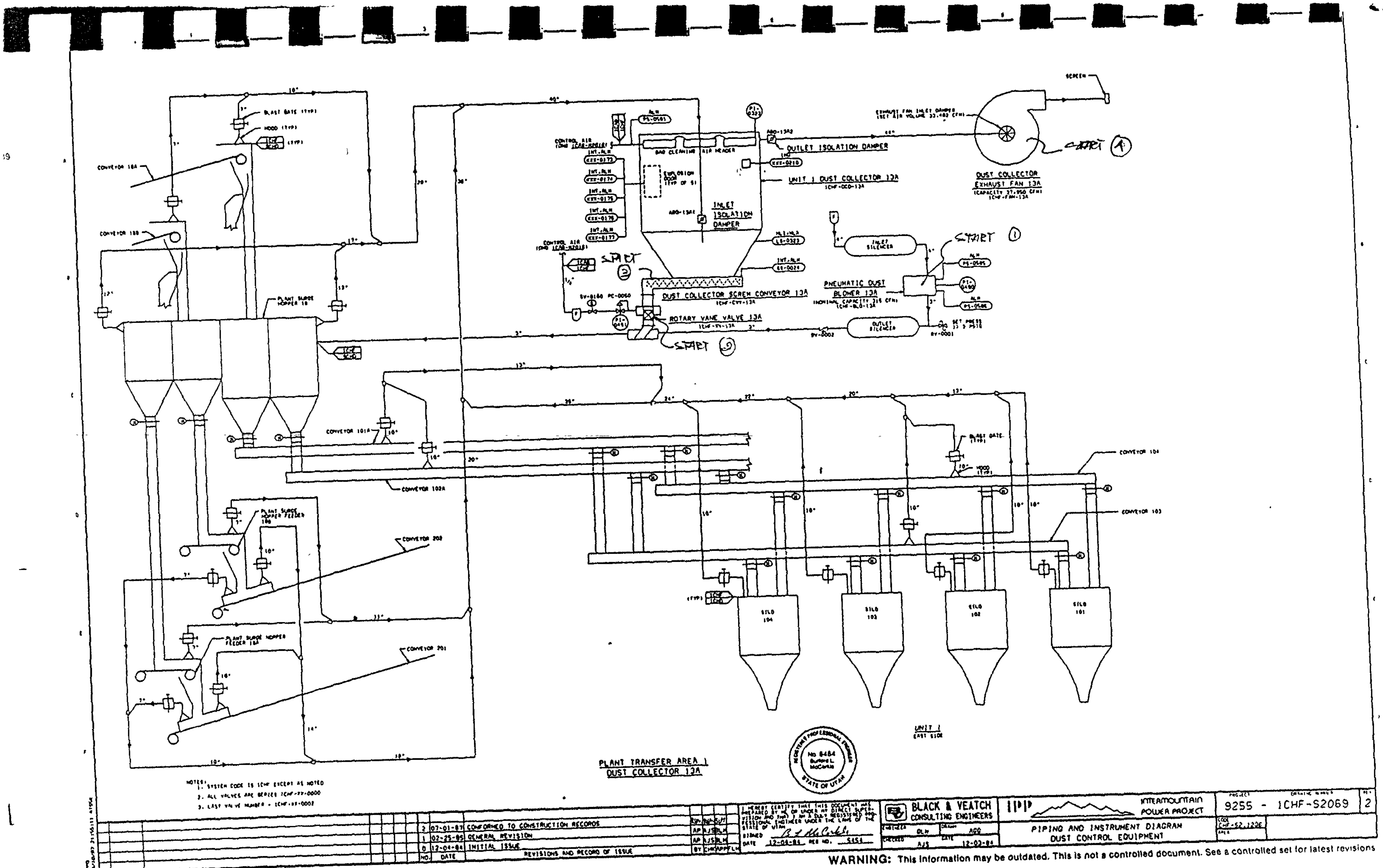
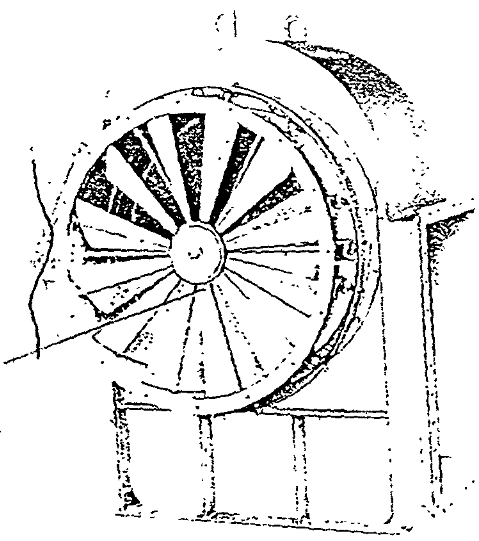


FIG 1 - DUST COLLECTOR 13A



Inlet dampers—external vane construction provides pre-spun air effect to reduce fan performance efficiently... not recommended for use with inlet box . . . maximum temperature: 750°F.

PLR

Flat backwardly inclined blade design for efficient air movement and minimum maintenance in contaminated air streams.

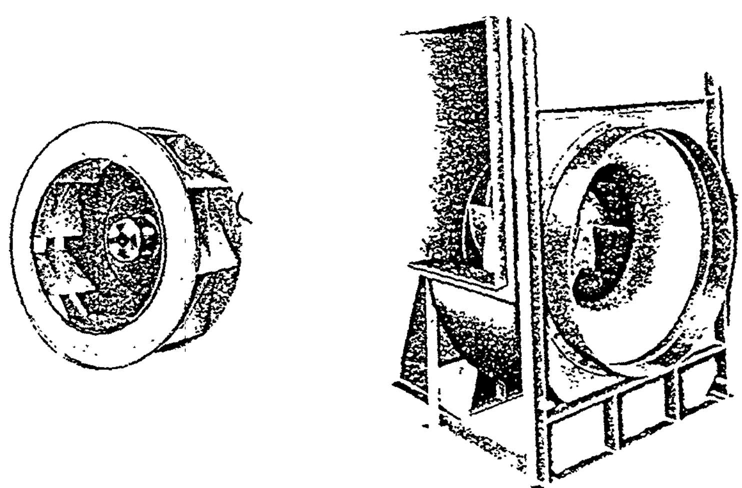


FIG. 3 1 CENTRIFUGAL FANS

FIG. 16 – DUST COLLECTOR MODELING

$T_{SP} = P_4 - P_3 - VP_3$

$T_{SP} = P_4 - P_1 - \Delta P_1 - \Delta P_2 - VP_3$

$T_{SP} = P_4 - VP_2 - (\Delta P_1 + \Delta P_3 + \Delta P_{DUCT})$

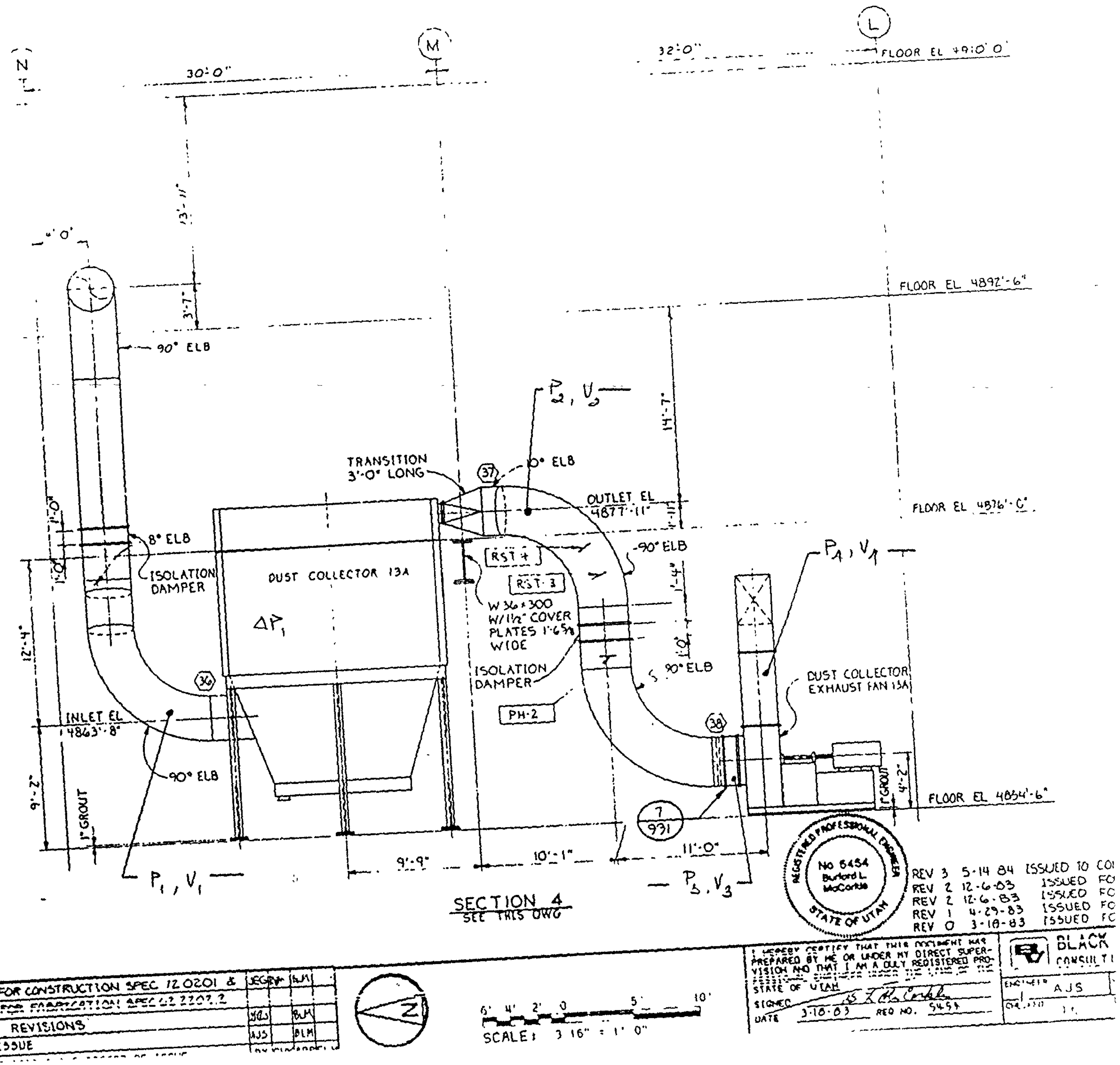
WHERE

$\Delta P_{DUCT, 13A}$ = 9.6" wg

ΔP_1 = -9" wg

P_4 = 2.6" wg

VP_3 = 1." wg



IP7_038978

3. Bag Cleaning System

Cyclic impulse air jet cleaning of filter bags is initiated by pressure differential switch. When the pressure drop across the filter bags reaches its set point (4" H20 to stop and 6" H20 to start), the pressure differential switch will initiate the cleaning cycle. The cleaning system will run for one (1) cycle (blow down every bag in the hopper) unless the high differential pressure (DP) still exists. If this happens, the system will go through another cycle. This will continue until the pressures are equal. If the DP is still greater than the set point after running through five (5) cycles, the DP's alarm will illuminate while the cleaning system continues running. As the fan shut down, the cleaning system will shut down automatically **after** ten (10) minutes. The cleaning cycle can also be initiated manually.

An adjustable timing sequence mechanism will automatically open and close solenoid valves, controlling the compressed airflow. A solenoid valve is furnished for each separately backblown tier of bags. Duration and frequency of the individual tier cleaning cycles will be regulated from an adjustable timing device capable of varying both duration and frequency of the cleaning operation.

Net Air-to-cloth ratio is defined as the airflow rates divided by the cloth area actually collecting dust at any time during collector operation, particularly during the filter cleaning process.

Filter bag and cleaning air data are listed as follows:

- Total Filtering Area: 2,984 sq ft
- Air-to-Cloth Ratio: 6.25:1
- Bag Cleaning Air Required at 100 Psig: 37.5 scfm
- Compressed Air Pressure Required: 100 Psig
- Differential Pressure to Stop/Initiate Cleaning: 4/6 inches gauge of H20

Refer to Table 2: Filter Bag And Cleaning Air Data

4. Exhaust Fans

Each dust collector is equipped with one induced draft type exhaust fan with the designed criteria as listed:

- New York Blower Model No: Class IV 40 PLR
- Size: 40 - 93.5% NW (Narrow Width)
- Wheel diameter: 40 inches
- Operating Speed: 1,780 rpm
- Wheel Tip Speed: 18,743 ft/sec
- Safe Maximum Speed: 1,945 rpm

- Critical Speed: 2,431 rpm
- Air Volume Flow Rate: 37,950 cfm
- Fan Static Pressure: 18.5 in H20
- U.S. Motor Horsepower/Voltage: 200/460
- Direct Drive Motor Speed: 1,800 rpm

Each fan is centrifugal type with backward inclined blades and is equipped with a radial vane type, external inlet damper with a manual positioning and locking device. Fan's wheel of dust collector 13A should be:

- Clockwise rotation reference from driven side.
- Vertical Up
- Damper is clockwise rotation.
- AMCA type B, spark resistant material (aluminum)

The fan is designed to have less than 1.5 mils operating vibration amplitude at any place as measured on the bearing housing of the fan and motor.

Refer to Table 3, Figures 3, 17 and 18.

5. Pneumatic Conveying Systems

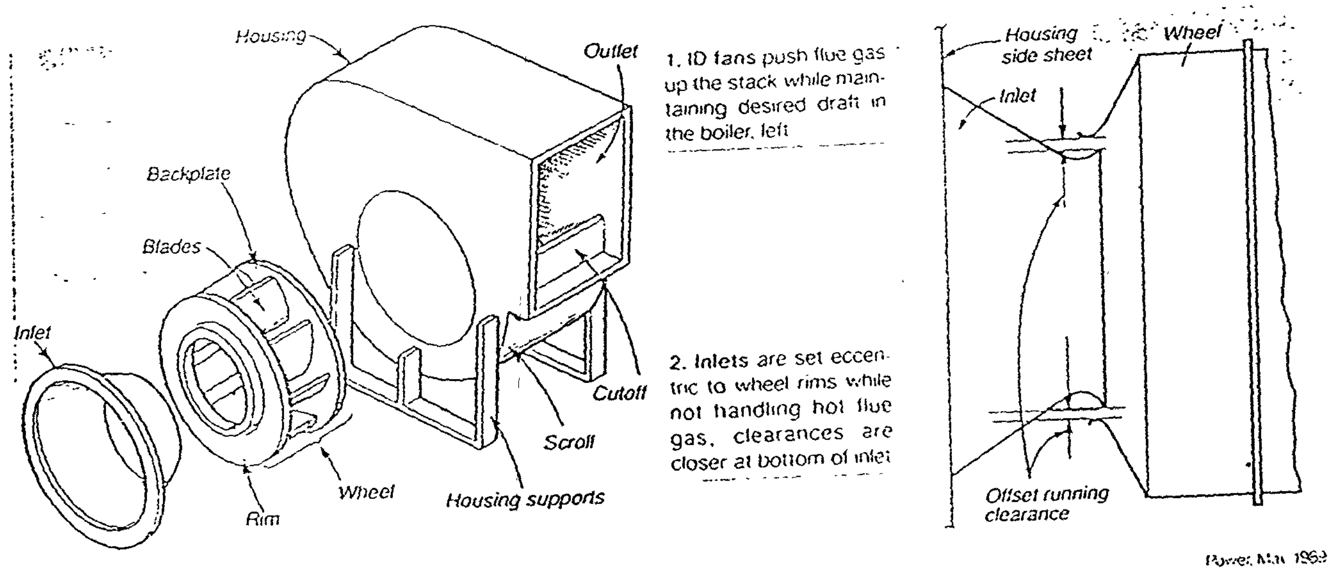
The pneumatic conveying systems consist of positive displacement blowers, rotary vane valves, and associated piping. The mechanical blowers convey the dust from the dust collectors to the dust return point. The blowers are equipped with Air Maze UM08HNB inlet filters, Universal inlet and outlet silencers, the necessary valves and pressure gages. The pneumatic dust transport blower for collector 13A data are:

- Model No: 5MVF
- Rotative Speed: 1,750 rpm
- Volume: 303 acfm
- Discharge Pressure: 3.6 Psig
- Motor HP/voltage: 15/460
- Smoot Relief Valve Model: PV3
- Relief Set Point: 9.2 Psig

6. Duct Work

The ductwork provides an enclosed passage for the collected dust and transport air for each of the individual collection systems. The duct work subcomponent consists of all straight ductwork, transitions, elbows, hooks, flexible connectors, cleanouts, dead-end caps, dampers, hangers and all other necessary accessories.

The duct work is fabricated from ASTM A53 Grade B standard weight pipe for sizes 24 inches and smaller. All duct work



70

FIGURA 4

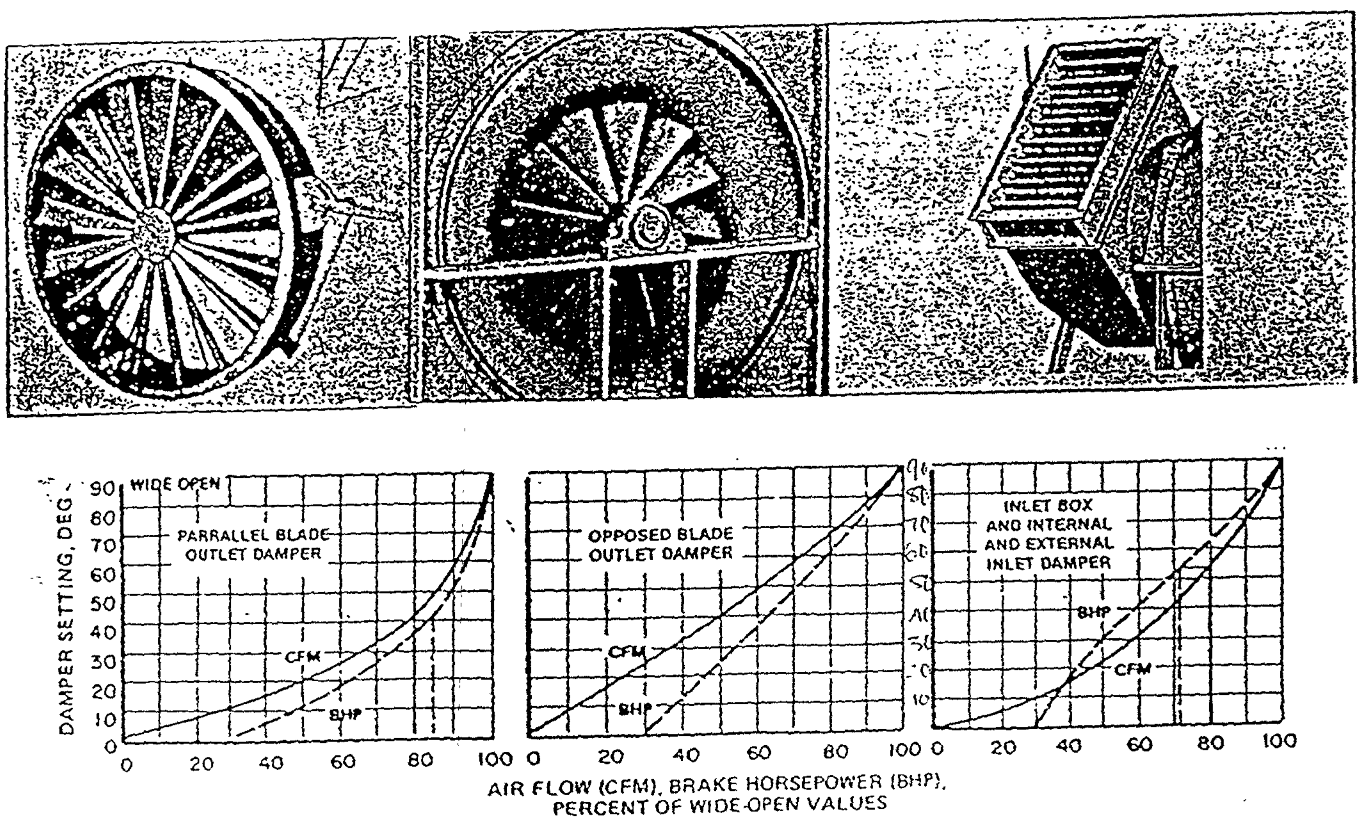


FIGURE 5

IP7_038981

19 "FAN ENGINEERING"
21 3-ring notebooks

FAN TESTING

Require Fans Models
submit to test for acceptance

Wheel min - 36" to 40"
size
for model

test must be made independently

DC diesel locomotive engine
1500 HP cutoff for most fan test facilities
duct 72"

(can test at lower speeds + take up in paper)

---

stress analysis testing
strain gauges ~$5K

FAN Diagnostics

Almost always operating on left side of curve
fan oversized   move to right of curve

reduce ΔP
open dampers (VAB, Bias inlet/outlet)
may have to waste air (bypass duct)

HOWDEN SIROCCO INC.
One Westinghouse Plaza
Hyde Park, MA 02136

TEST SUMMARY (RAW DATA)

# SIDE	TP IN "WG	VP IN "WG	SP IN "WG	SP OUT	BAR.	TO	TW	TIN	TOUT	RPM
DRIVE	-.619	.4502	-1.578		25.41	72	54	67.3		898
OUTBOARD	-.463	.376	-.803	37.05	25.40	64	51	67.7	99.8	897
DRIVE		.4414	-.9903		25.41	65	46	64.8		1194
OUTBOARD	-.4837	.376	-.6958	36.53	25.40	63	51	68.17	114.7	
DRIVE	-.5113	.4671	-.6156		25.44	65	46	66.83		
OUTBOARD	-.586	.4766	-.7895	37.05	25.45	65	46	66.4	93.6	897
DRIVE	-.579	.4565	-.6022		25.47	70	49	75.2		
OUTBOARD	-.593	.447	-.749	36.734	25.46	69	49	69.4	118.6	

TEST SUMMARY (RESULTS CONVERTED TO CURVE CONDITION)

LOW SPEED FAN 2A
287515 CFM 34.64"WG 55.5° VANES 2241 HP

HIGH SPEED FAN 2A
285496 CFM 33.98"WG 75° VANES 4116 HP

LOW SPEED FAN 2B
306219 CFM 34.44"WG 54.5° VANES 2238 HP

HIGH SPEED FAN 2B
300566 CFM 34.2678"WG 74.5° VANES 4187 HP

C. Ireland

TEST 1

RIVE SIDE

$T_d = 72$ $T_W = 54$ BAR $= 25.41$ FROM AMCA 203 FIG. N-1 $\rho_0 = 0.0631$

UTBOARD SIDE

$T_d = 64$ $T_W = 51$ BAR $= 25.4$ $\rho_0 = 0.0641$

RIVE SIDE

$$\rho_3 = \rho_0 \left(\frac{P_{s3} + 13.6\,P_b}{13.6\,P_b}\right)\left(\frac{td_0 + 460}{td_3 + 460}\right) \qquad \frac{T_3\,\rho_3}{P_3} = \frac{T_0\,\rho_0}{P_0}$$

$$= 0.0631 \left(\frac{-1.578 + 13.6 \times 25.41}{13.6 \times 25.41}\right)\left(\frac{72 + 460}{67.3 + 460}\right)$$

$$= 0.0631 \times .9954 \times 1.0089$$

$$= 0.06337\ \text{lb/ft}^3$$

Traverse Area $= 51.45\ \text{ft}^2$

$$V_3 = 1096\sqrt{\frac{P_{v3}}{\rho_3}} = 1096\sqrt{\frac{.4502}{.06337}}$$

$$= 2921\ \text{fpm}$$

$$Q_3 = 2921 \times 51.45$$

$$Q_1 = Q_3 = 150304\ \text{cfm.}$$

OUTBOARD SIDE

$$\rho_3 = 0.0641\left(\frac{-.803 + 13.6 \times 25.4}{13.6 \times 25.4}\right)\left(\frac{64 + 460}{67.7 + 460}\right)$$

$$= 0.0641 \times 0.9977 \times 0.99299 = 0.063504\ \text{lb/ft}^3$$

Traverse Area $= 51.45\ \text{ft}^2$

$$V_3 = 1096\sqrt{\frac{.376}{.063504}} = 2666.88\ \text{fpm}$$

$$Q_3 = 2666.88 \times 51.45 = 137211\ \text{cfm.}$$

$$Q_1 = Q_3$$

Fan Static Pressure $= P_{S_2} - P_{T_1}$

$= 37.05 - \left(\frac{.611 + .463}{2}\right)$

$=$ [redacted] 37.591

CONVERT TO CURVE CONDITIONS

TEST SPEED = CURVE SPEED

TEST SIZE = CURVE SIZE

TEST DENSITY $\simeq$ 0.0636

Curve Density $\simeq$ 0.0586

Curve Flow = 287,515

Curve Pressure $=$ [redacted] $37.591 \times \frac{.0586}{.0636}$

$=$ [redacted]

$= 34.64"$ Wg

VANES AT 57% OPEN
or 34.5°

ST 2

DRIVE SIDE

$T_d = 65$ $T_w = 46$ BAR $= 25.41$ $\rho_0 = 0.06405$

OUTBOARD SIDE

$T_d = 63$ $T_w = 51$ BAR $= 25.41$ $\rho_0 = 0.0642$

DRIVE SIDE

$$\rho_3 = \rho_0 \left(\frac{P_{s_3} + 13.6 P_b}{13.6 P_b}\right)\left(\frac{td_0 + 460}{td_3 + 460}\right)$$
$$= 0.06405 \left(\frac{-.9903 + 13.6 \times 25.41}{13.6 \times 25.41}\right)\left(\frac{65+460}{64.8+460}\right)$$
$$= 0.06405 \times .9971 \times 1.0004$$
$$= 0.06389 \text{ lb/ft}^3$$

Traverse Area $= 51.45 ft^2$

$$V_3 = 1096\sqrt{\frac{P_{v_3}}{\rho_3}} = 1096\sqrt{\frac{.4414}{.06389}}$$
$$Q_3 = 2880.78 \times 51.45$$
$$Q_1 = Q_3 = 148216 \text{ cfm}$$

OUTBOARD SIDE

$$\rho_3 = \rho_0 \left(\frac{P_{s_3} + 13.6 P_b}{13.6 P_b}\right)\left(\frac{td_0 + 460}{td_3 + 460}\right)$$
$$= 0.0642 \left(\frac{-.6959 + 13.6 \times 25.41}{13.6 \times 25.41}\right)\left(\frac{63+460}{68.2+460}\right)$$
$$= 0.0642 \times .9980 \times .990155$$
$$= 0.06344 \text{ lb/ft}^3$$

Traverse Area $= 51.45 ft^2$

$$V_3 = 1096\sqrt{\frac{P_{v_3}}{\rho_3}} = 1096\sqrt{\frac{.376}{.06344}}$$
$$= 2668.2 \text{ cfm}.$$
$$Q_1 = Q_3 = 2668.2 \times 51.45 = 137,280.3 \text{ cfm}$$

Total flow at Inlet = 285496 cfm

2

Fan static Pressure $= P_{S_2} - P_{T_1}$

$= 36.53 - (-.4837)$

$=$ [redacted] 37.0137

Convert to Curve.

Curve Flow = 285496 CFM

Curve Pressure = [redacted] 37.0137 $\times \dfrac{.0584}{\text{[redacted] } .0636}$

$=$ [redacted] 33.987 at 15° open

Test 3

Drive Side & Outboard Side

$T_d = 65$ $T_w = 46$ $P_b = 25.445$ $\rho_o = .06425$

... values.

$$\rho_3 = \rho_o \left(\frac{P_{s_2} + 13.6 P_b}{13.6 P_b}\right)\left(\frac{td_o + 460}{td_3 + 460}\right)$$

$$= .06425\left(\frac{-.69 + 13.6 \times 25.445}{13.6 \times 25.445}\right)\left(\frac{65+460}{66.2+460}\right)$$

$$= .06425 \times 0.998 \times .9977$$

$$= 0.063975$$

...rive side $V_3 = 1096\sqrt{\frac{P_{V_3}}{\rho_3}} = 1096\sqrt{\frac{.4671}{.064}}$

$$V_3 = 2960.9 \text{ fpm}$$

$$Q_1 = Q_3 = 2960.9 \times 51.45 = 152,339 \text{ cfm}$$

...let side $V_3 = 1096\sqrt{\frac{.4766}{.064}}$

$$= 2991 \text{ fpm}$$

$$Q_1 = Q_3 = 153,880$$

Total Inlet flow = 306219 cfm

Fan Static Pressure = $P_{s_2} - P_{T_1} = 37.05 - (-.54865)$
$= 37.598$

Convert to Curve

Curve flow = 306219

Curve Press = $37.598 \times \frac{.0586}{.063975}$

$= 34.44$

at 53.3% open or 35.5° from shut.

IP7_038990

Test 4

Drive Side & Outboard Side

$T_D = 70$ $T_W = 49$ $P_b = 25.465$ $\rho_0 = .0637$

$$\rho_3 = \rho_0\left(\frac{-.585 + 13.6 P_b}{13.6 P_b}\right)\left(\frac{70+460}{72+460}\right)$$

$$= .0637 \times .9983 \times .9962$$

$$= .0636$$

Drive Side $V_3 = 1096\sqrt{\frac{.4565}{.0636}} = 2936.31$ fpm

$Q_1 = Q_3 = 151,073$ cfm

Out Side $V_3 = 1096\sqrt{\frac{.447}{.0636}} = 2906$ fpm

$Q_1 = Q_3 = 149,393$ cfm

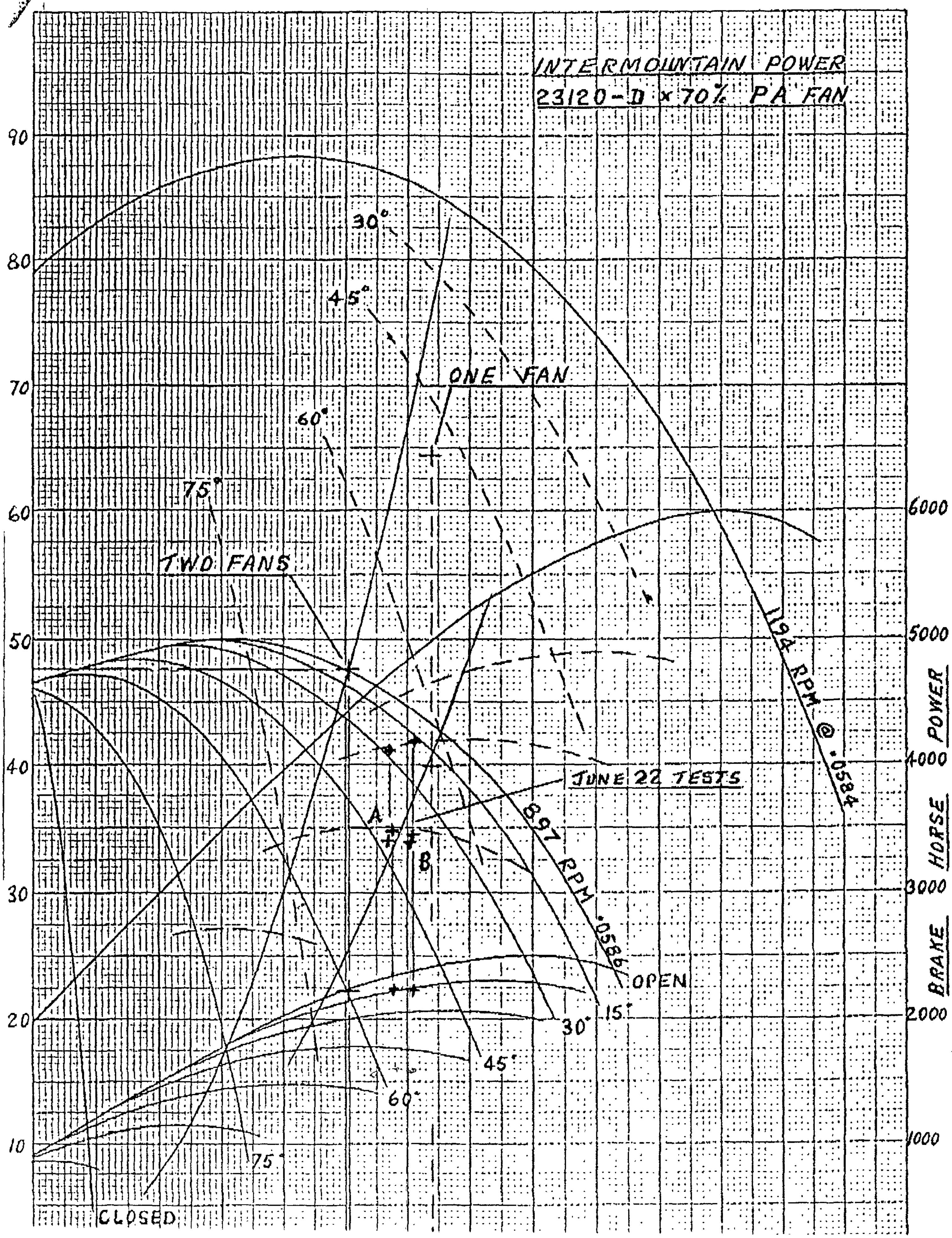
Total Inlet flow = 300,566.09 cfm.

Fan static Pressure = $P_{S_2} - P_{T_1} = 36.734 - (-.585)$

$= 37.319.$

To Curve $37.319 \times \left(\frac{.0584}{.0636}\right) = 34.2678$

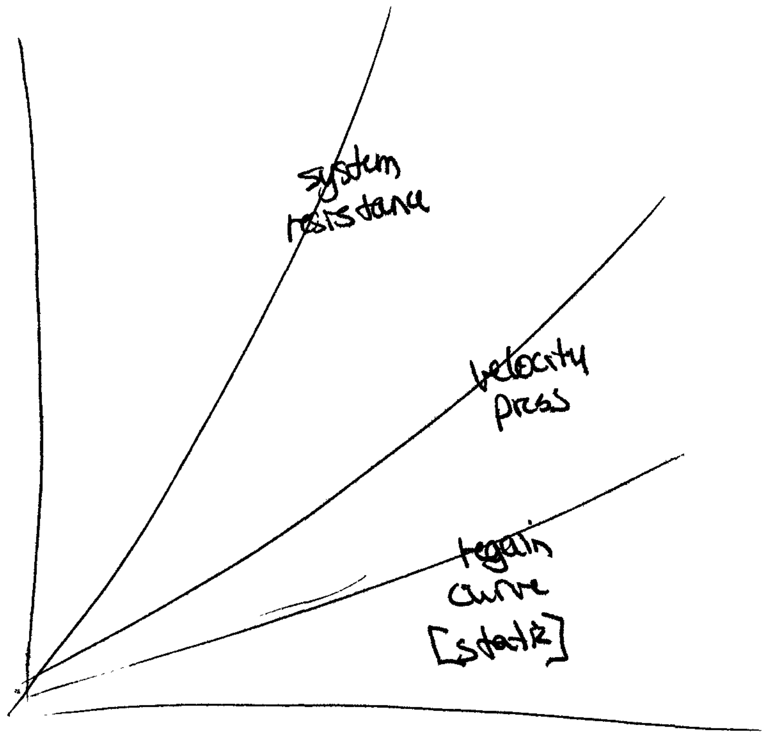
at 15.5° from shut ~ 31.1% open


INTERMOUNTAIN POWER
23120-D × 70% PA FAN
90
80
70
60
50
40
30
20
10
6000
5000
4000
3000
2000
1000
BRAKE HORSE POWER
30°
45°
60°
75°
ONE FAN
TWO FANS
1194 RPM @ .0584
897 RPM .0586
JUNE 22 TESTS
A
B
OPEN
15°
30°
45°
60°
75°
CLOSED

To Do

JC PA fan curves final (2) speed Hi/Low
2/1 fan operation

Projects modal fan blades axial
CT FD



regain curve + velocity press = total press
press

Variable Pitch

↑ maint

Variable Speed Drives

power comes down w/ speed control

INTERMOUNTAIN GEN. STATION
B & V PROJECT 9255
B & V FILE 61.0405.05-90956

required at fan des. = 33,000 ACFM

PERFORMANCE CURVE

nyb | The New York Blower Company

To determine performance at another RPM multiply:

1- CFM x K
2- SP x K^2
3- BHP x K^3

where K is new RPM divided by RPM shown at right.

Customer's No. 36XX-20690-20707-40
Customer FLEXKLEEN CORP.
Tagging SEE BELOW (narrow width wheel)
Size 40-93.5%NW Type PLR CL IV WHEEL
37950 CFM at 18.5 "SP at 0.0768 D. (density)
at 1780 RPM at 147.6 BHP
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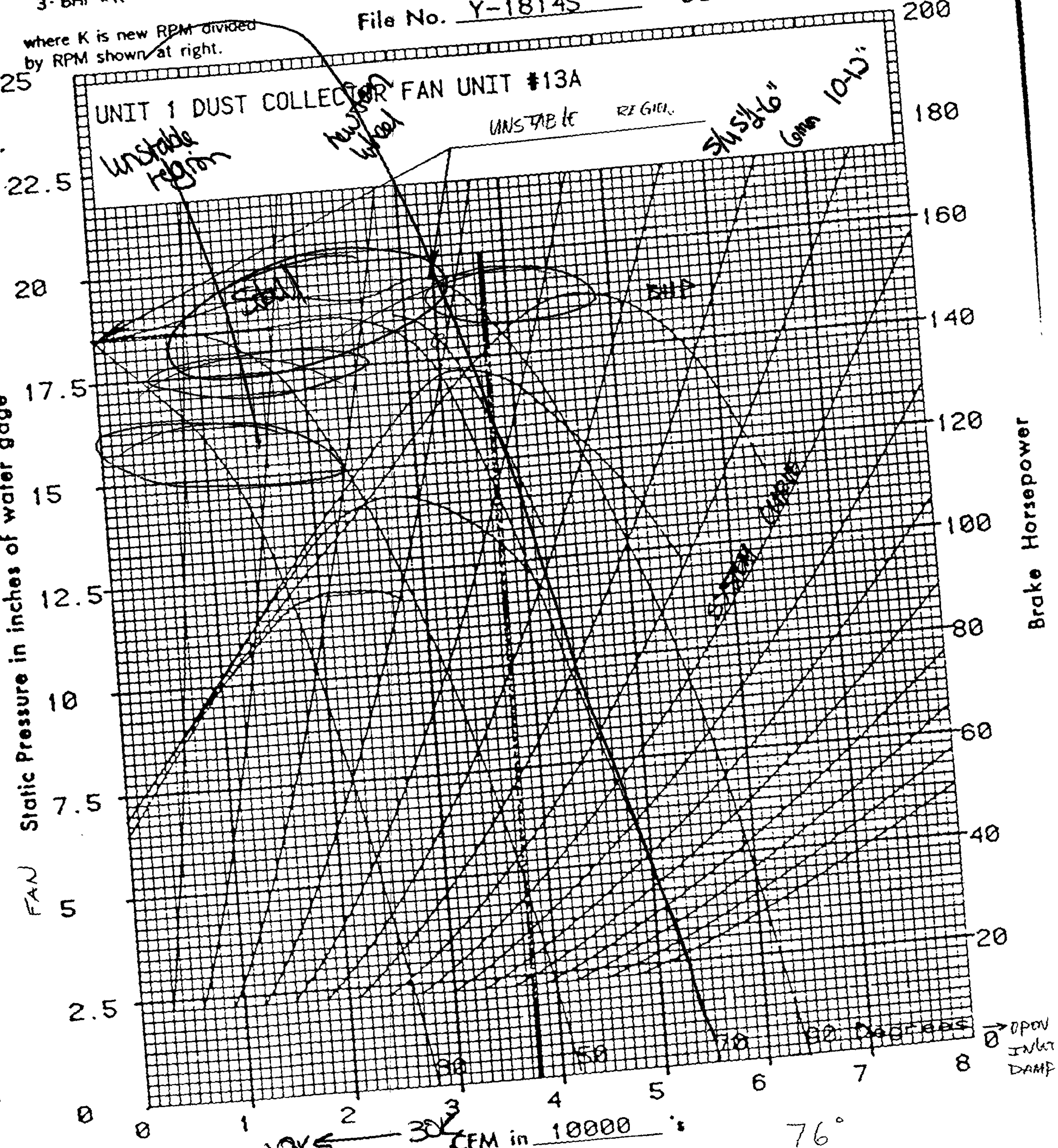


FIG. 11

IP7_038995

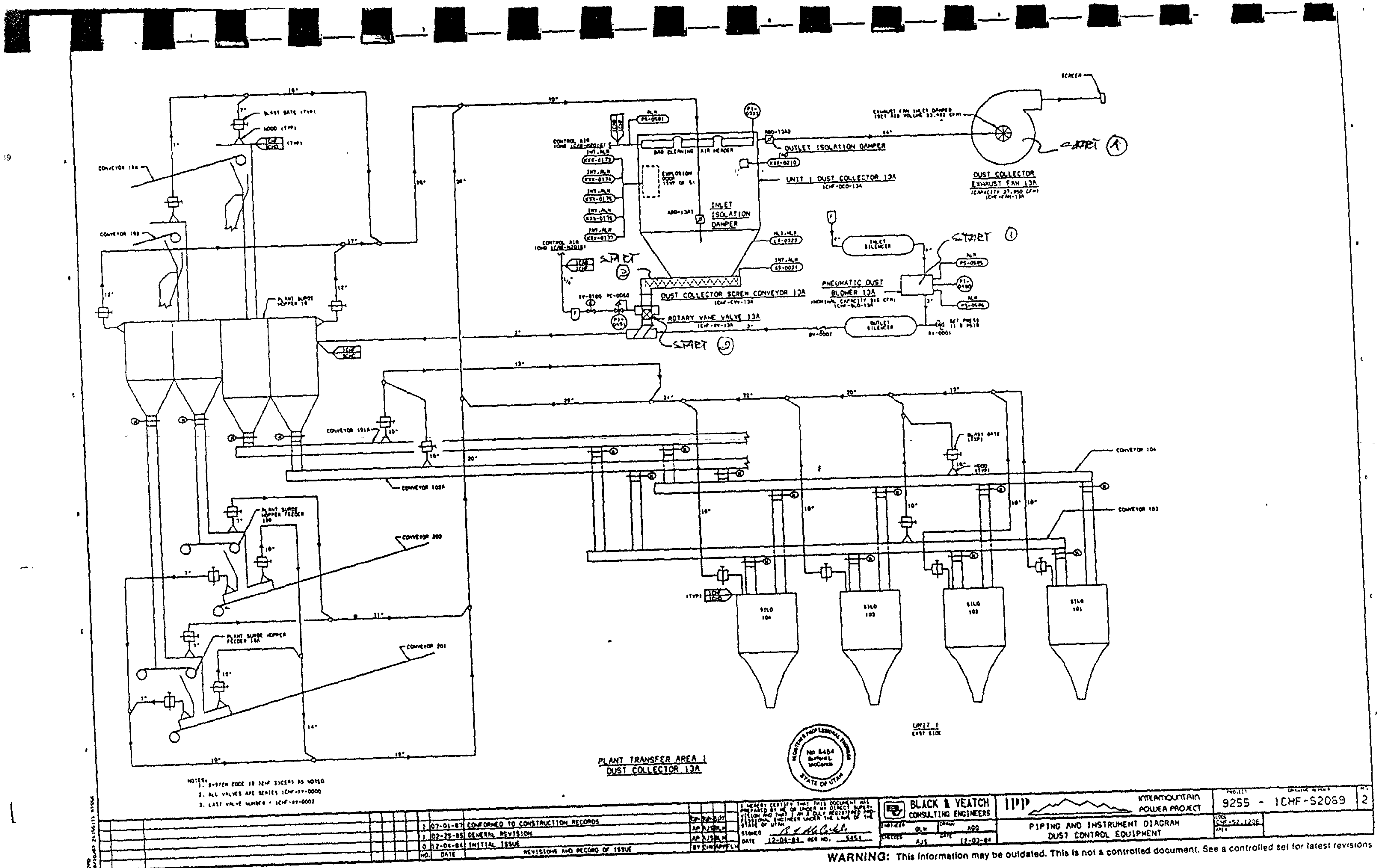
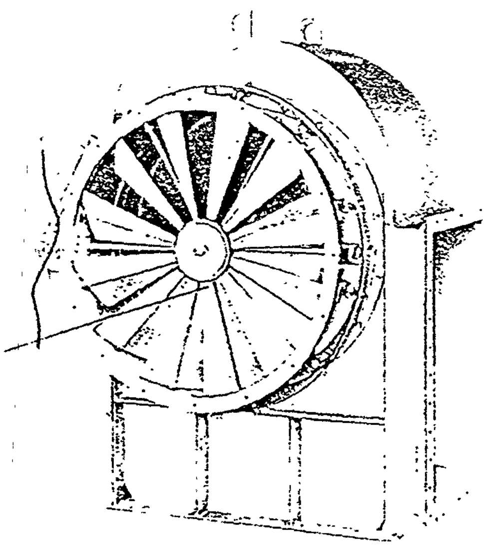


FIG 1 - DUST COLLECTOR 13A

IP7_038996

Inlet dampers—external vane construction provides pre-spun air effect to reduce fan performance efficiently... not recommended for use with inlet box . . . maximum temperature: 750°F.



PLR

Flat backwardly inclined blade design for efficient air movement and minimum maintenance in contaminated air streams.

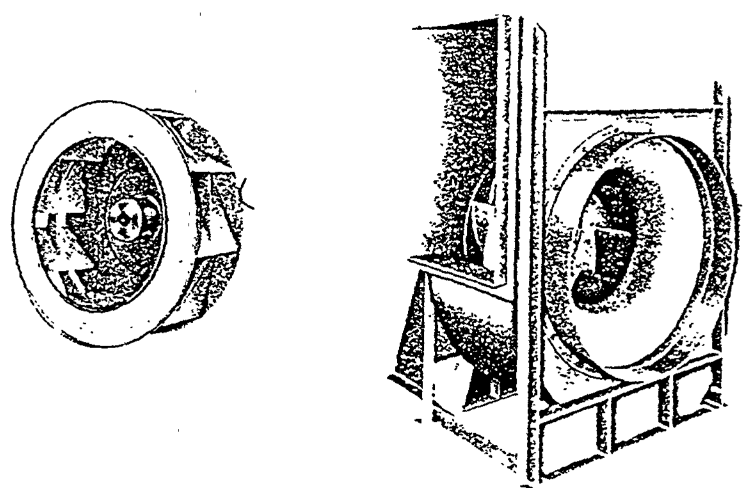


FIG. 3 1 CENTRIFUGAL FANS

FIG. 16 – DUST COLLECTOR MODELING

$T_{SP} = P_4 - P_2 - VP_3$

$T_{SP} = P_4 - P_1 - \Delta P_1 - \Delta P_2 - VP_3$

$T_{SP} = P_4 - VP_3 - (\Delta P_1 + \Delta P_2 + \Delta P_{DUCT})$

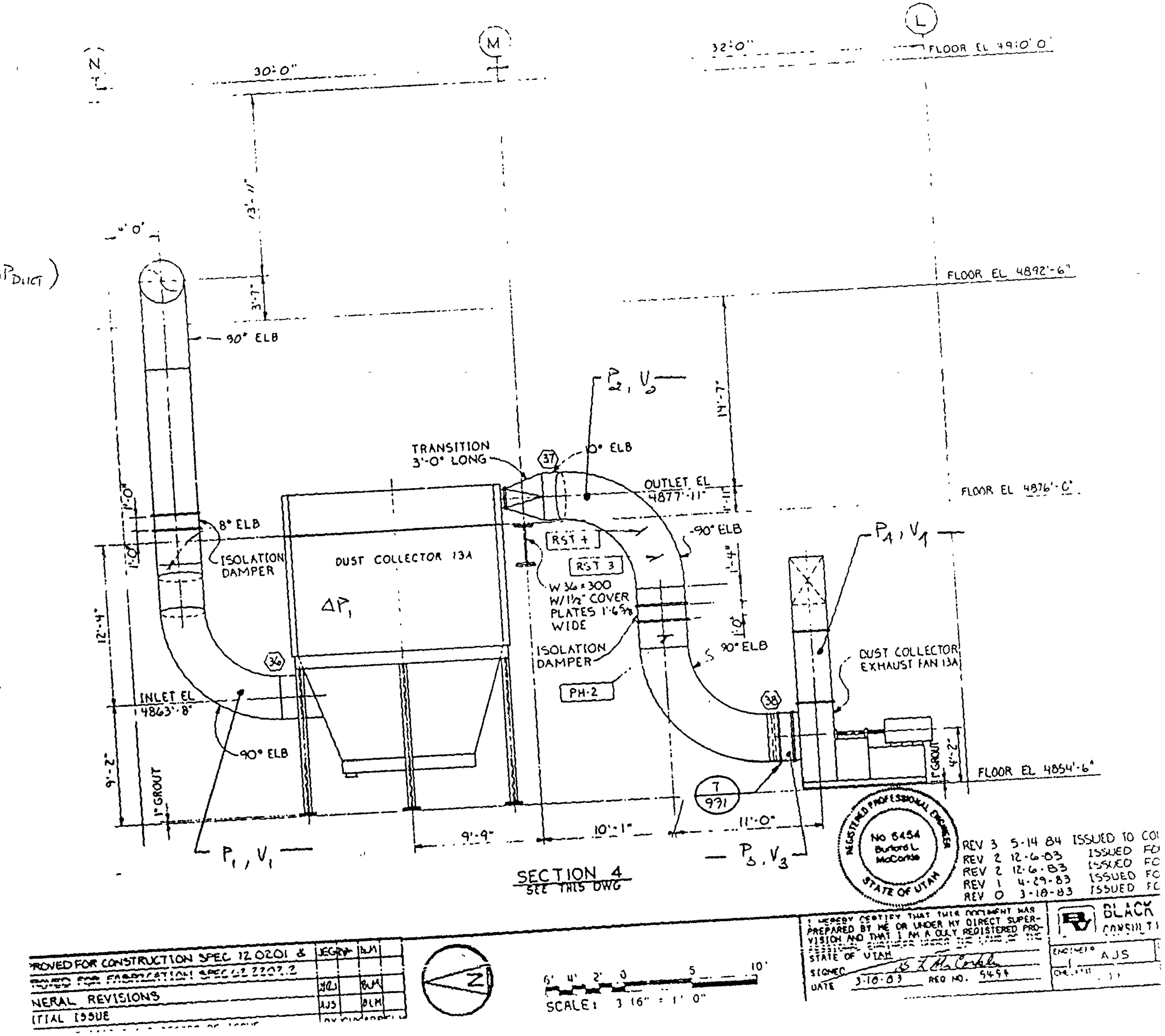
WHERE

$\Delta P_{DUCT,\ 13A} = 9.6''\ wg$

$\Delta P_1 = -9''\ wg$

$P_4 = 2.6''\ ws$

$VP_3 = 1.''\ ws$



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- Differential Pressure to Stop/Initiate Cleaning: 4/6 inches gauge of H20

Refer to Table 2: Filter Bag And Cleaning Air Data

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- Size: 40 - 93.5% NW (Narrow Width)
- Wheel diameter: 40 inches
- Operating Speed: 1,780 rpm
- Wheel Tip Speed: 18,743 ft/sec
- Safe Maximum Speed: 1,945 rpm

- Critical Speed: 2,431 rpm
- Air Volume Flow Rate: 37,950 cfm
- Fan Static Pressure: 18.5 in H20
- U.S. Motor Horsepower/Voltage: 200/460
- Direct Drive Motor Speed: 1,800 rpm

Each fan is centrifugal type with backward inclined blades and is equipped with a radial vane type, external inlet damper with a manual positioning and locking device. Fan's wheel of dust collector 13A should be:

- Clockwise rotation reference from driven side.
- Vertical Up
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- AMCA type B, spark resistant material (aluminum)

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- Model No: 5MVF
- Rotative Speed: 1,750 rpm
- Volume: 303 acfm
- Discharge Pressure: 3.6 Psig
- Motor HP/voltage: 15/460
- Smoot Relief Valve Model: PV3
- Relief Set Point: 9.2 Psig

6. Duct Work

The ductwork provides an enclosed passage for the collected dust and transport air for each of the individual collection systems. The duct work subcomponent consists of all straight ductwork, transitions, elbows, hooks, flexible connectors, cleanouts, dead-end caps, dampers, hangers and all other necessary accessories.

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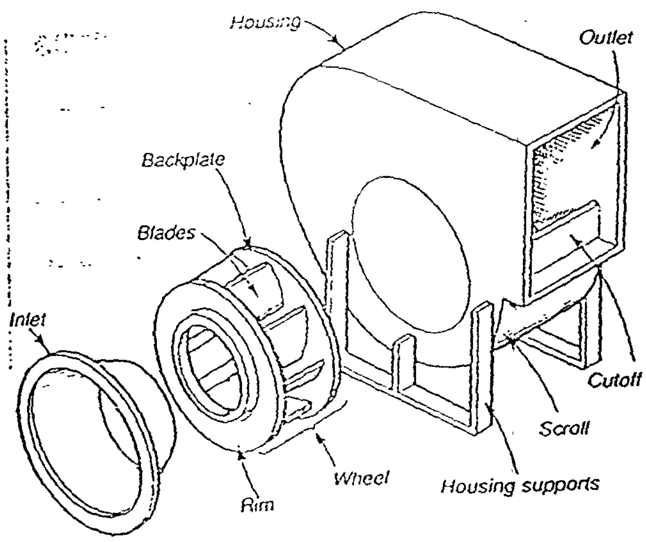
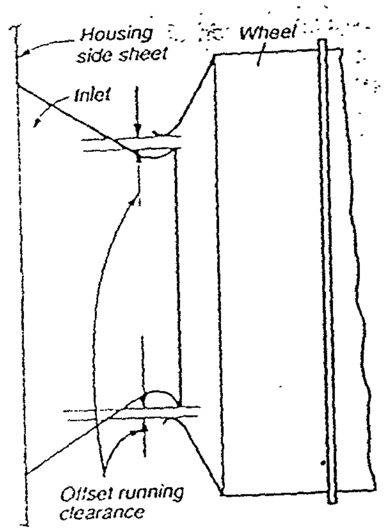


1. ID fans push flue gas up the stack while maintaining desired draft in the boiler, left

2. Inlets are set eccentric to wheel rims while not handling hot flue gas, clearances are closer at bottom of inlet



Power, May 1982

FIGURA 4

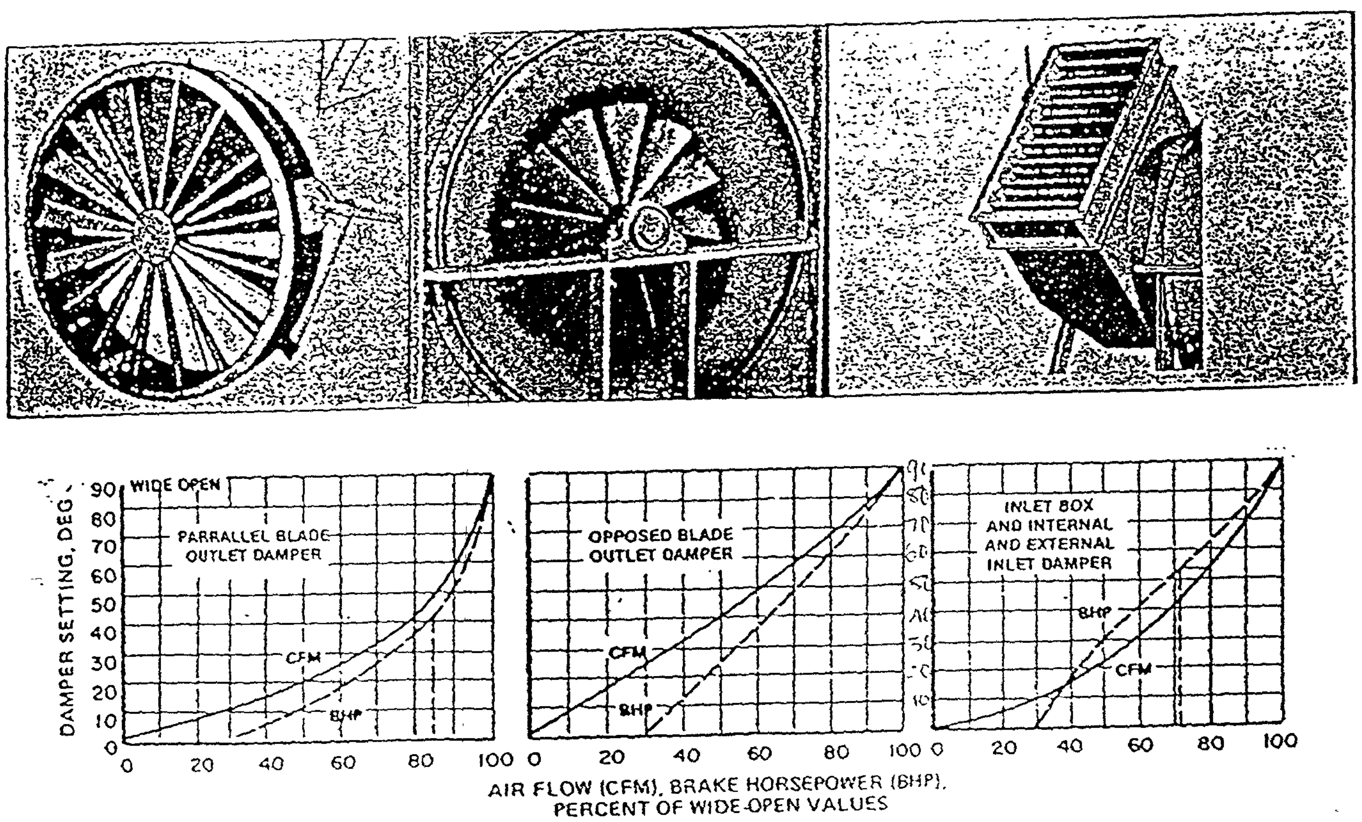
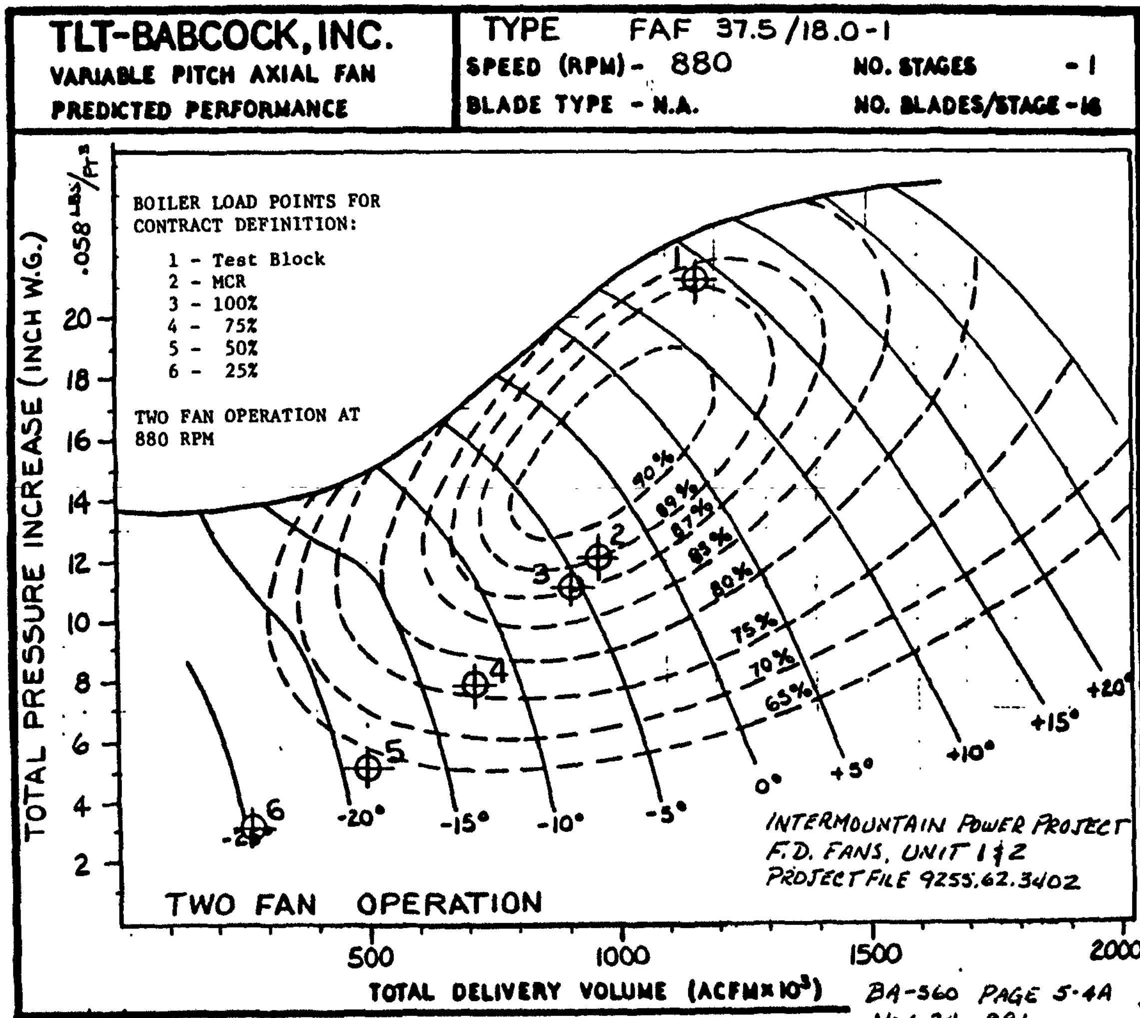


FIGURE 5

TABLE 3-1. FORCED DRAFT FAN PREDICTED PERFORMANCE

TLT Variable Pitch Axial fan (16 blade) [2]
2speed 880/700rpm (4305/2018 HP)

Item	Load Point: Test Block	MCR	100	75	50	25
Inlet Air Temperature, F	110	110	110	110	110	110
Inlet Air Density, lb/ft^3	0.0580	0.0580	0.0580	0.0580	0.0580	0.0580
Capacity, each fan						
Pounds per hour	4,018,300	3,335,600	3,135,800	2,461,800	1,712,200	904,100
Actual cfm	1,154,700	958,500	901,100	707,400	492,000	259,800
Fan Static Pressure, in. wg	19.5	10.9	10.0	7.2	4.8	3.0
Fan Static Efficiency, per cent	84.0	89.7	90.0	90.0	76.5	55.0
Design Fan Speed, rpm	894	717	717	717	717	717
Input Horsepower	4,305	2,018	1,730	966	520	227



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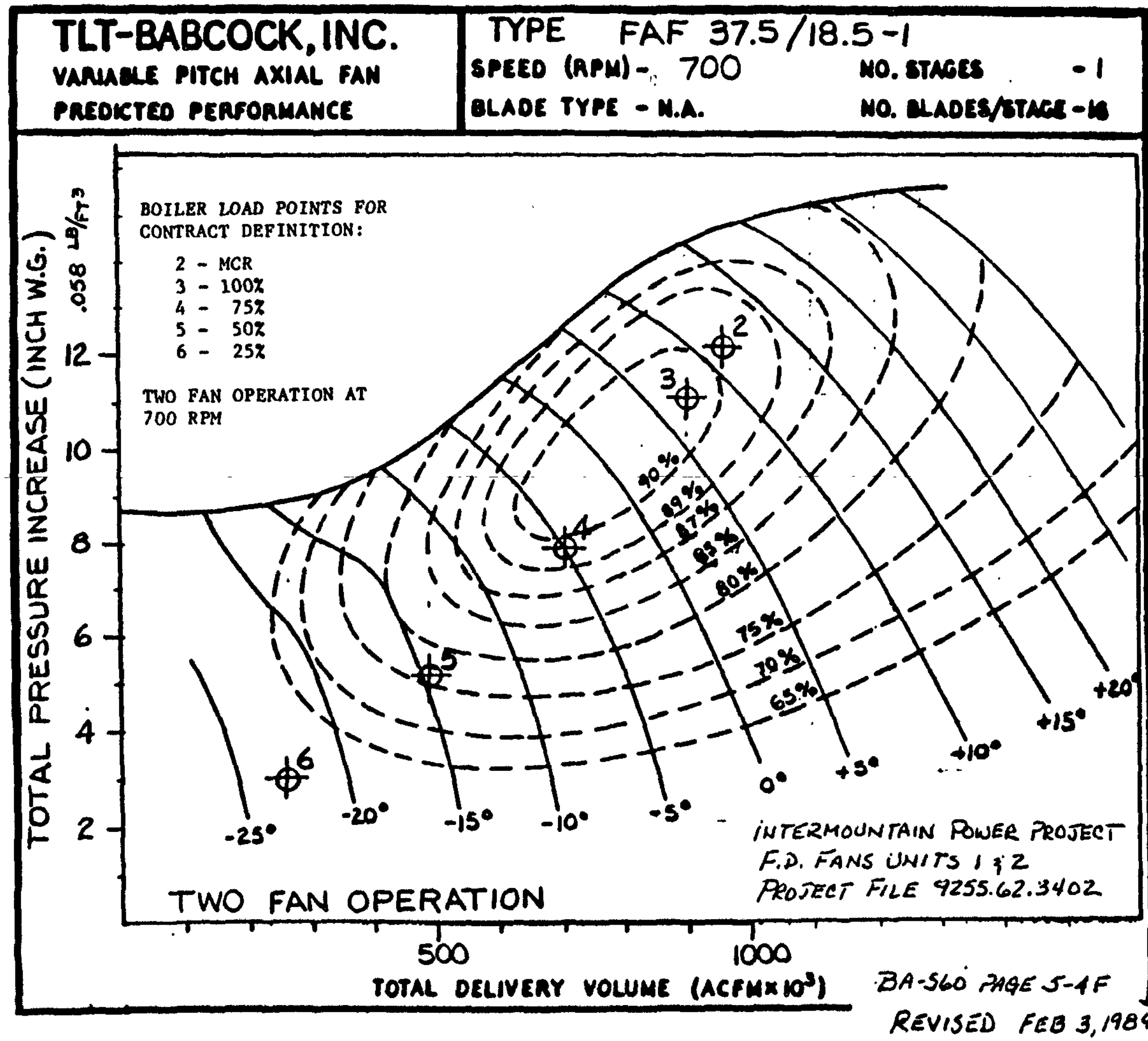
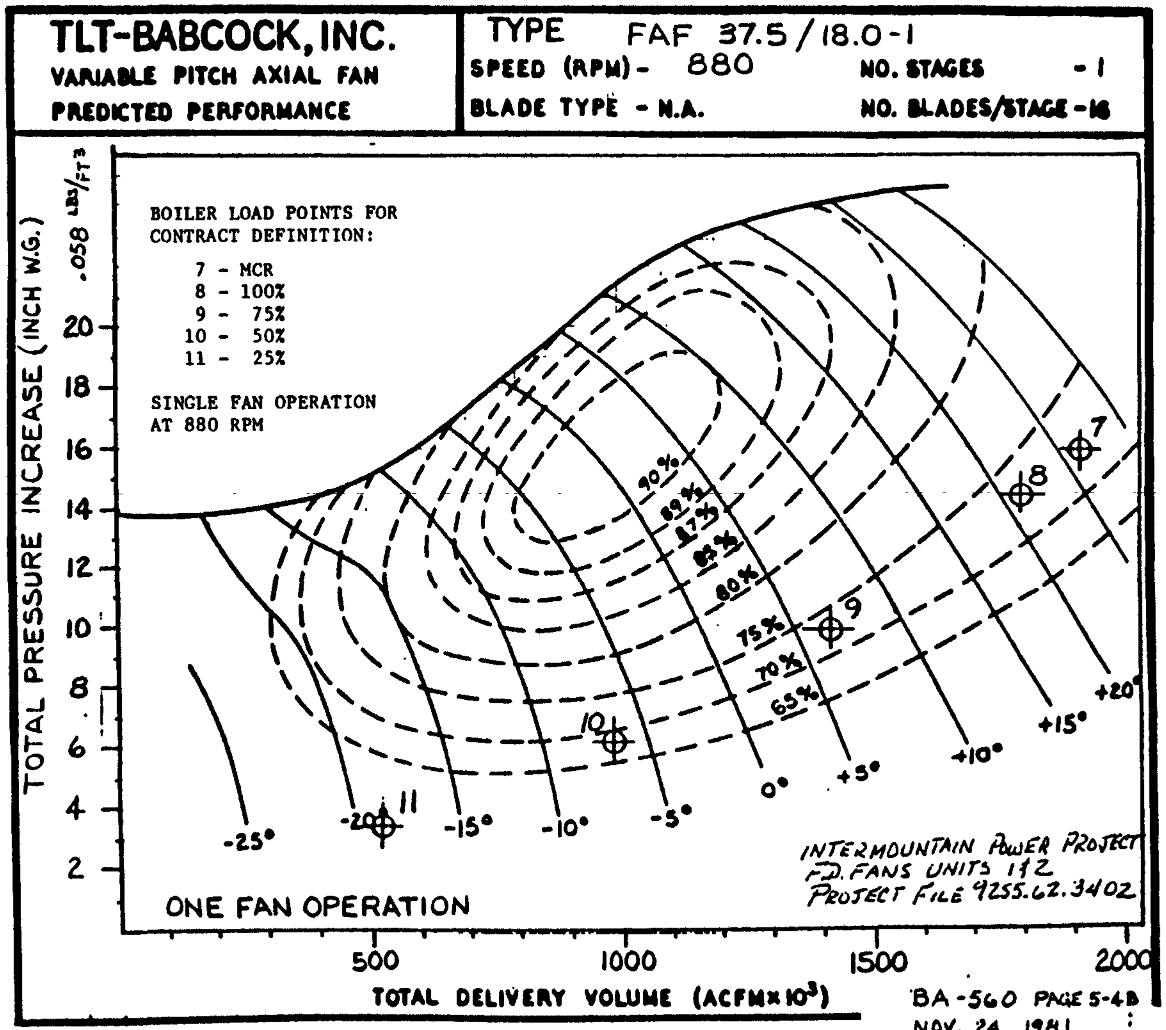
NOV 24, 1981
REVISED FEB 3, 1984

FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
880 RPM--TWO FAN OPERATION
FIGURE 3-1

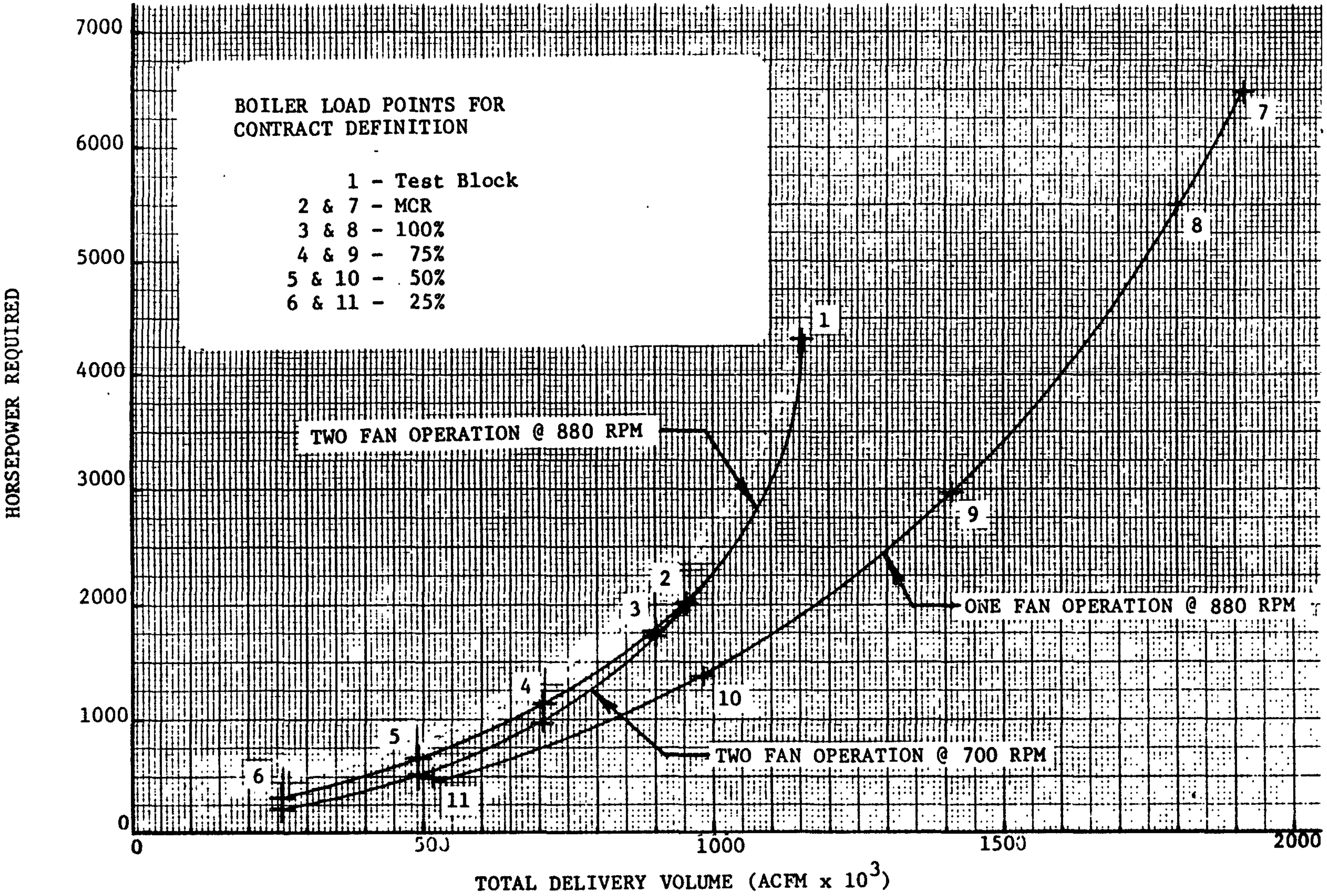


FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
700 RPM--TWO FAN OPERATION
FIGURE 3-2



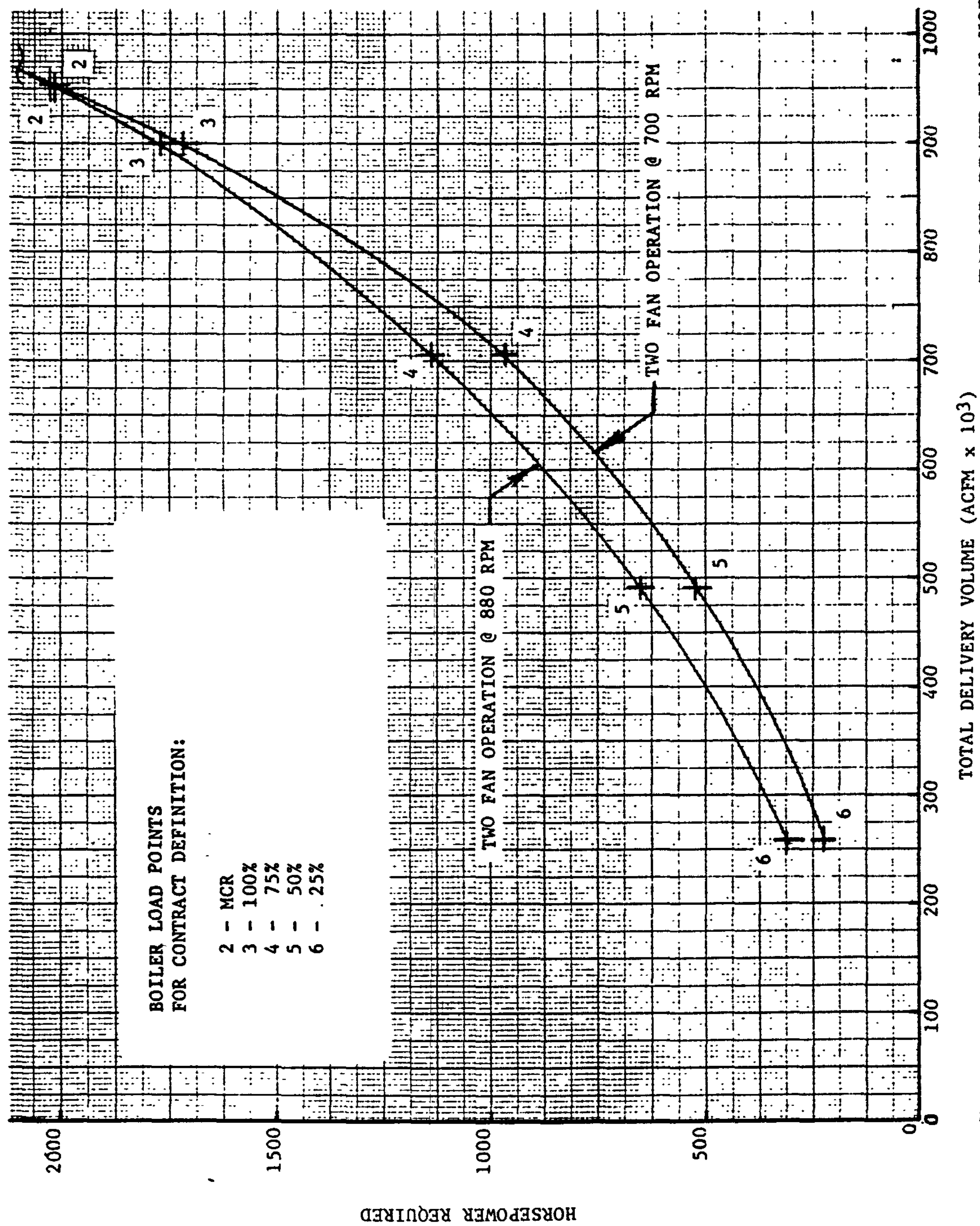
BA-560 PAGE 5-4B
NOV. 24, 1981
REVISED FEB 3, 1984

FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
880 RPM--SINGLE FAN OPERATION
FIGURE 3-3



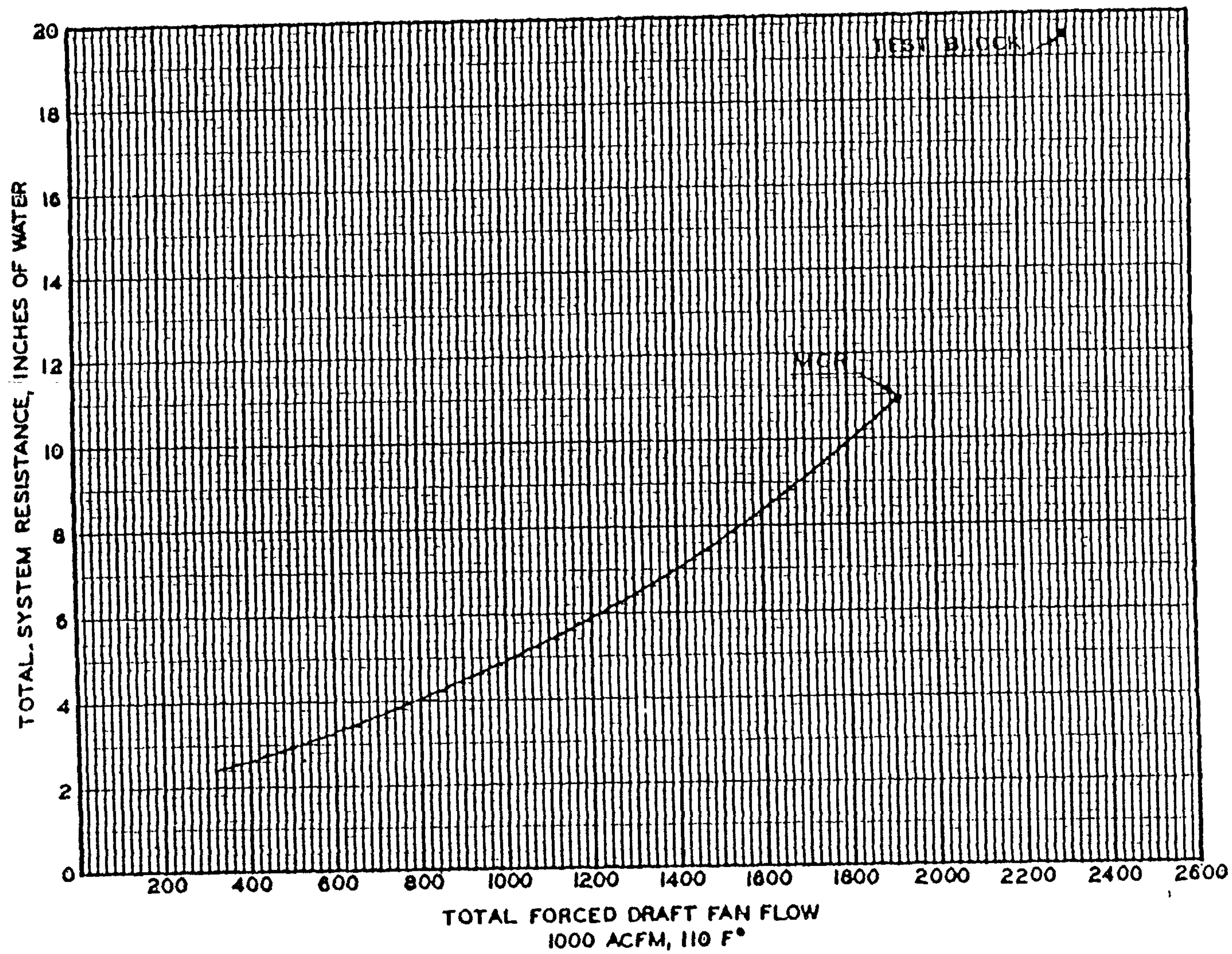
SHOP DRAWING NO. 9255.3402.06-10004

FORCED DRAFT FAN HORSEPOWER
PERFORMANCE CURVE
FIGURE 3-4



SHOP DRAWING NO. 9255.3402.06-10006

FORCED DRAFT FAN HORSEPOWER
PERFORMANCE CURVE--
OPERATING LOAD POINTS
FIGURE 3-5



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PROJECT
9255

IPP INTERMOUNTAIN POWER PROJECT

FORCED DRAFT FANS SYSTEM CURVE

DM-0032

IP7_039008

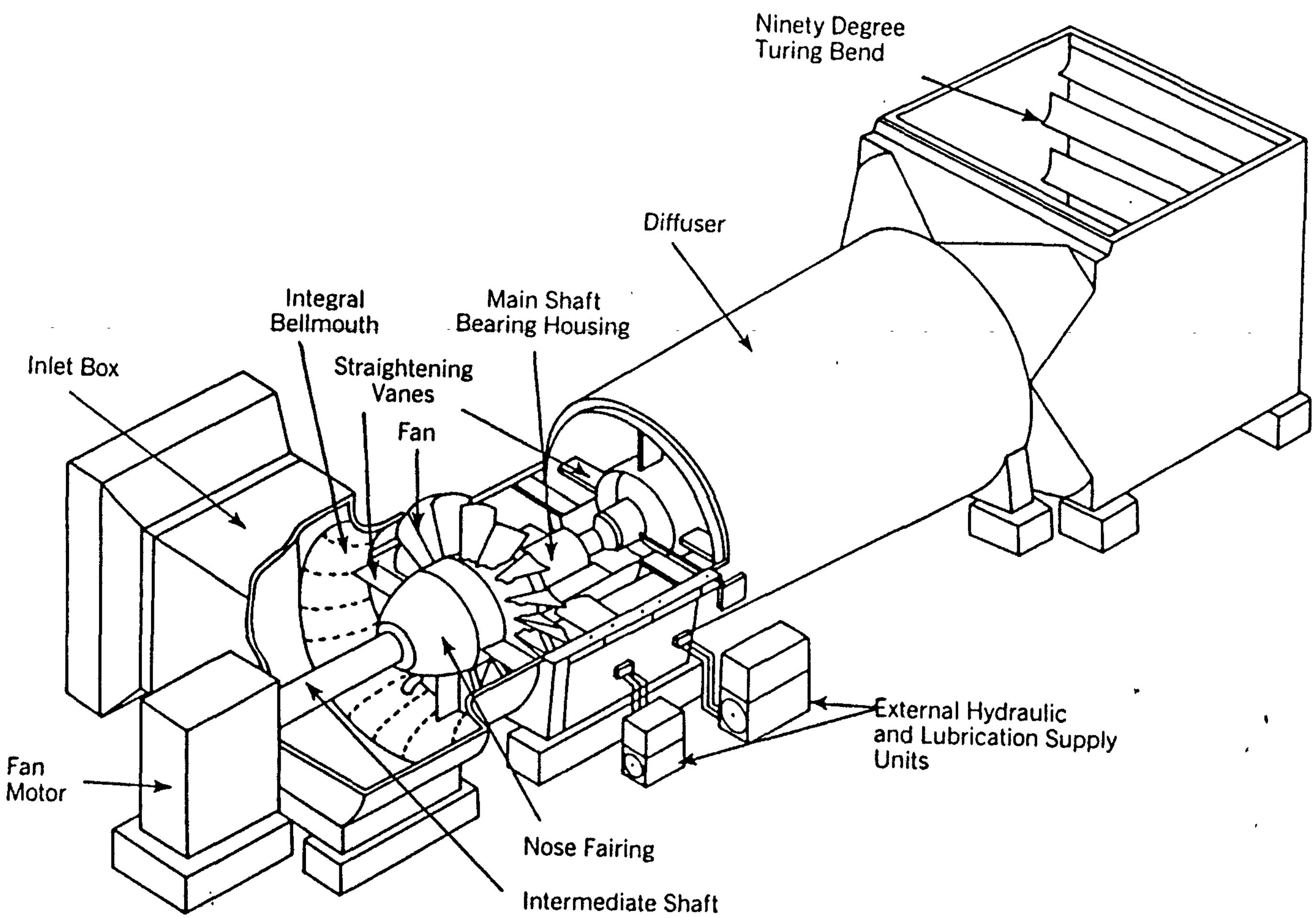


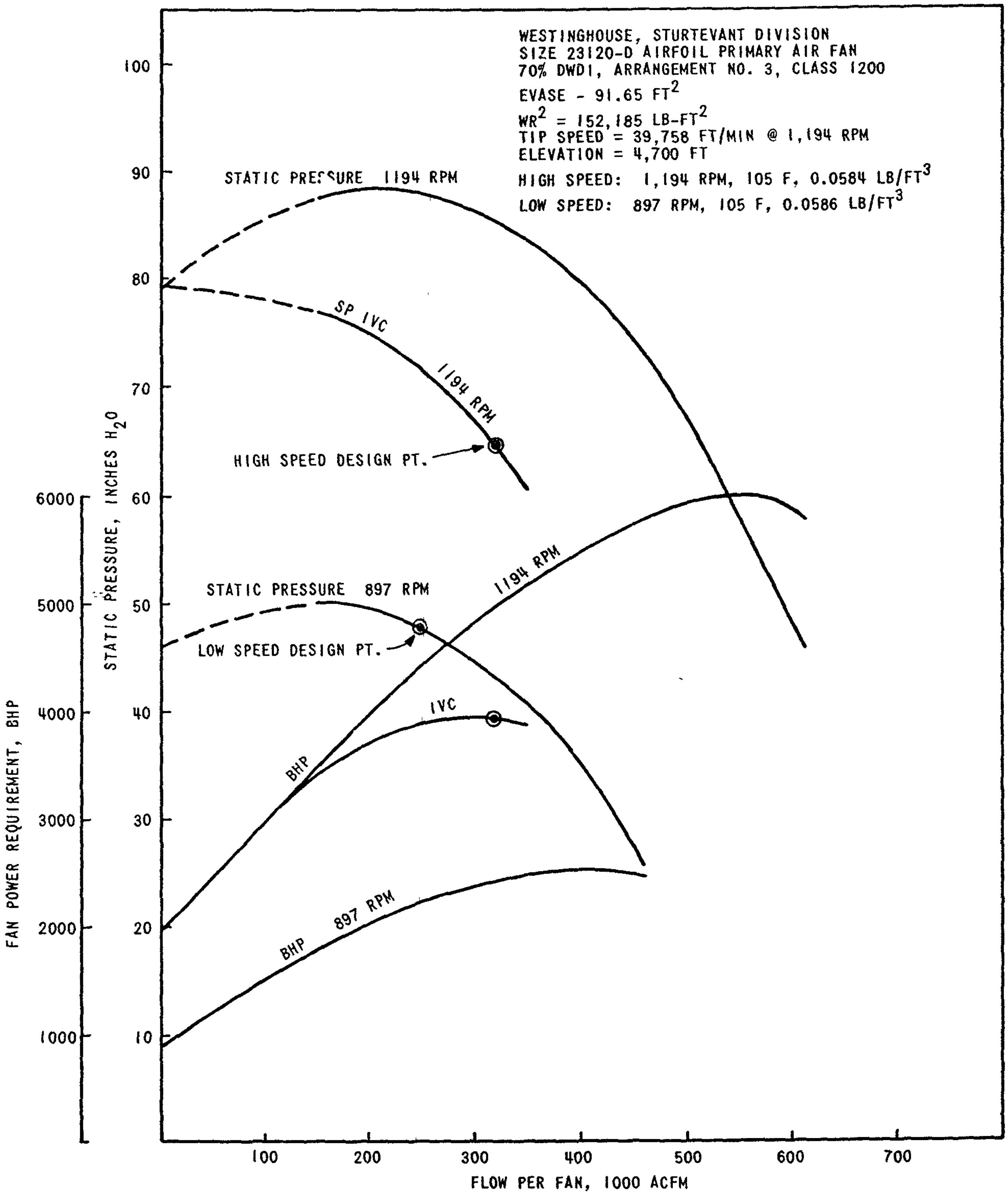
Figure 1

TLT- AXIAL FAN
FORCED DRAFT (2)

IP7_039009

TABLE 3-2. PRIMARY AIR FAN PREDICTED PERFORMANCE
Westinghouse-Sturtevant Centrifugal airfoil [2]
2 speed (1194/897 rpm) (3810/2061 HP)

Item	Test Block	MCR
Inlet Air Temperature, F	105	105
Inlet Air Density, lb/ft^3	0.0588	0.0588
Capacity, each fan		
Pounds per hour	1,120,300	882,000
Actual cfm	317,500	250,000
Fan Static Pressure, in. wg	62.5	44.5
Fan Static Efficiency, per cent	81.9	84.9
Design Fan Speed, rpm	1,194	897
Input Horsepower	3,810	2,061



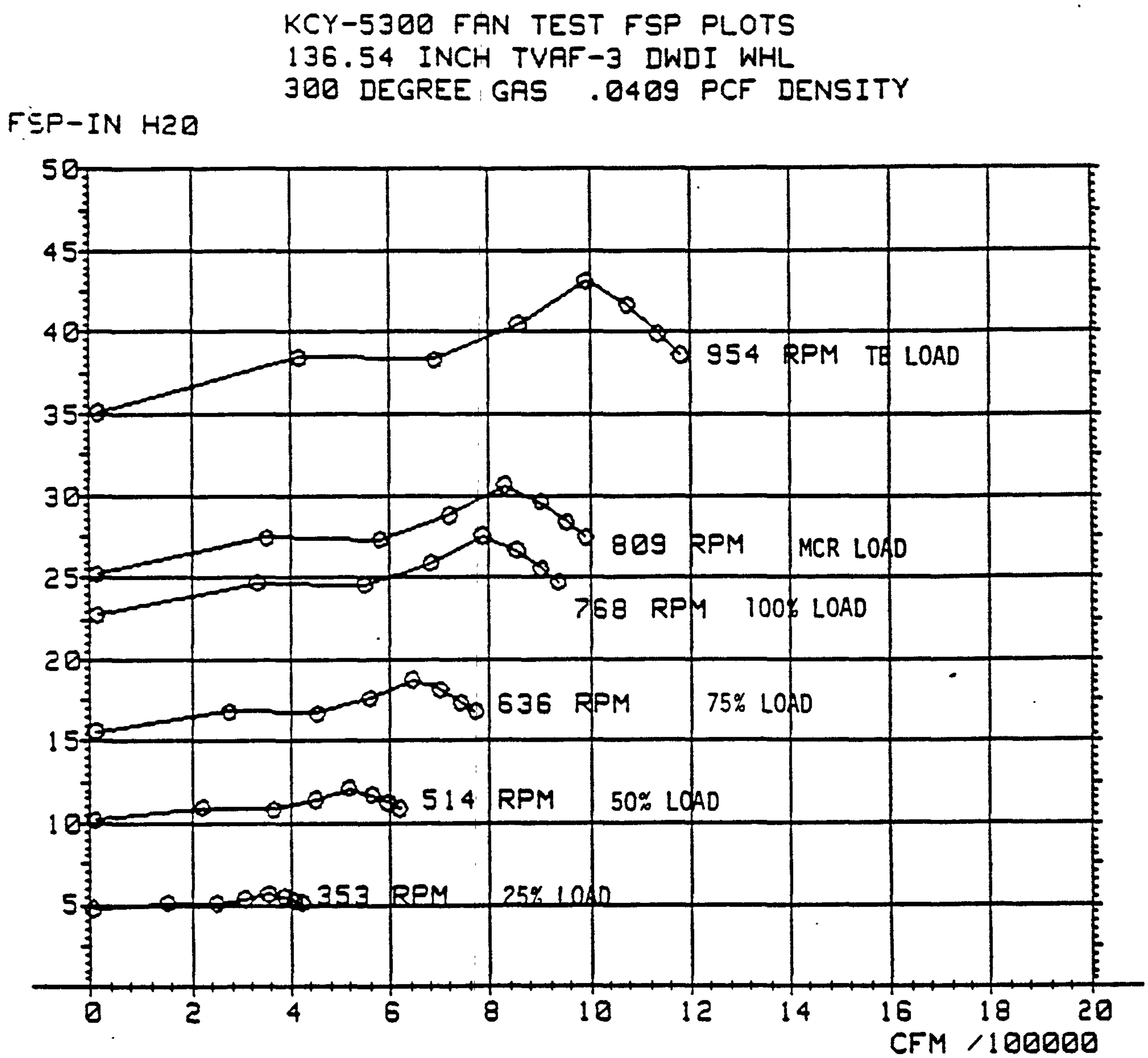
PRIMARY AIR FAN STATIC
PRESSURE AND HORSEPOWER
PERFORMANCE CURVES
FIGURE 3-6

ID fans - centrifugal (airfoil, double width, double inlet) 4 ~ 25%
adjustable speed brushless synchronous motor - 7415 HP (W)

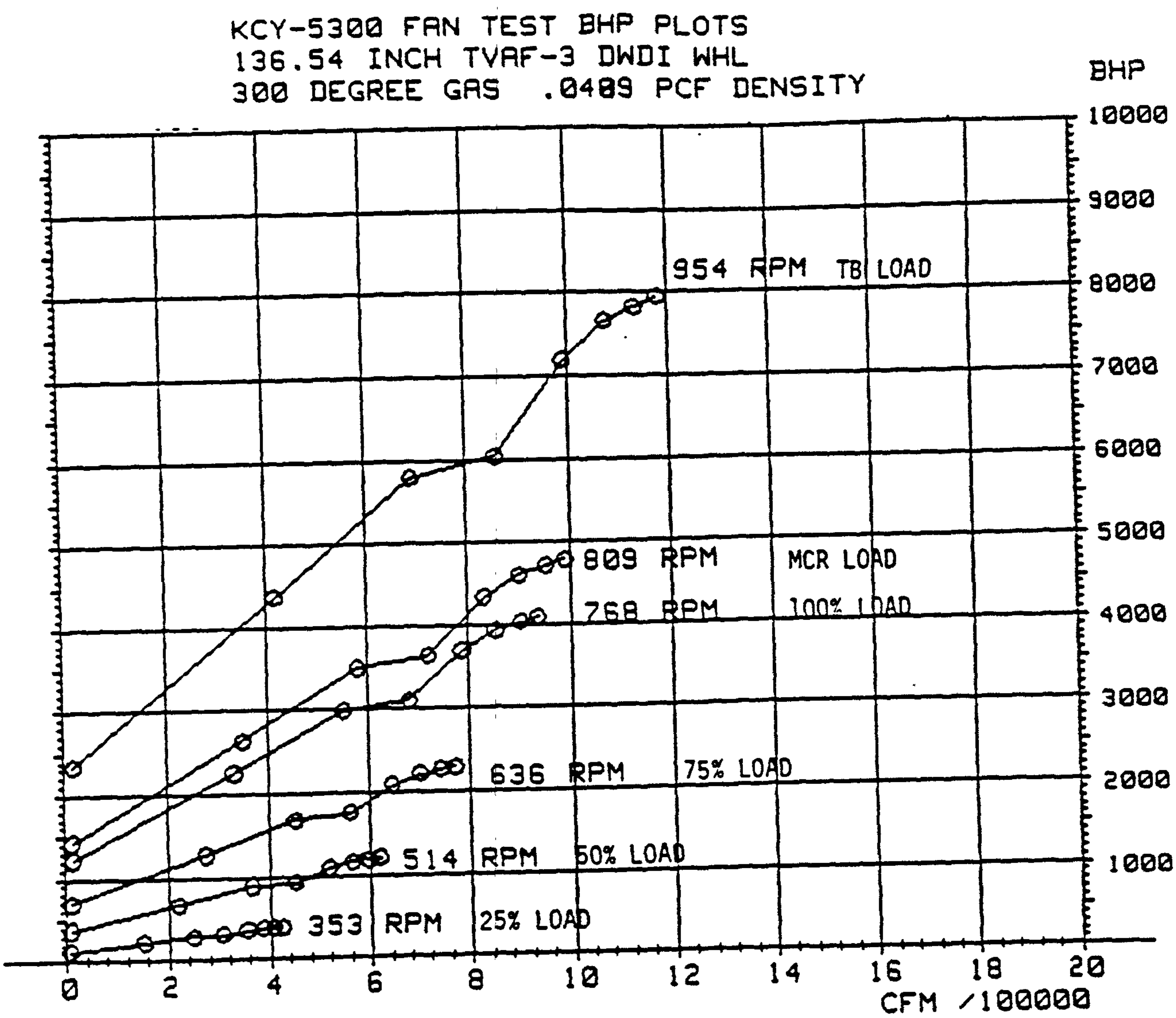
TABLE 3-1. INDUCED DRAFT FAN DESIGN CONDITIONS

Westinghouse - Sturtevant

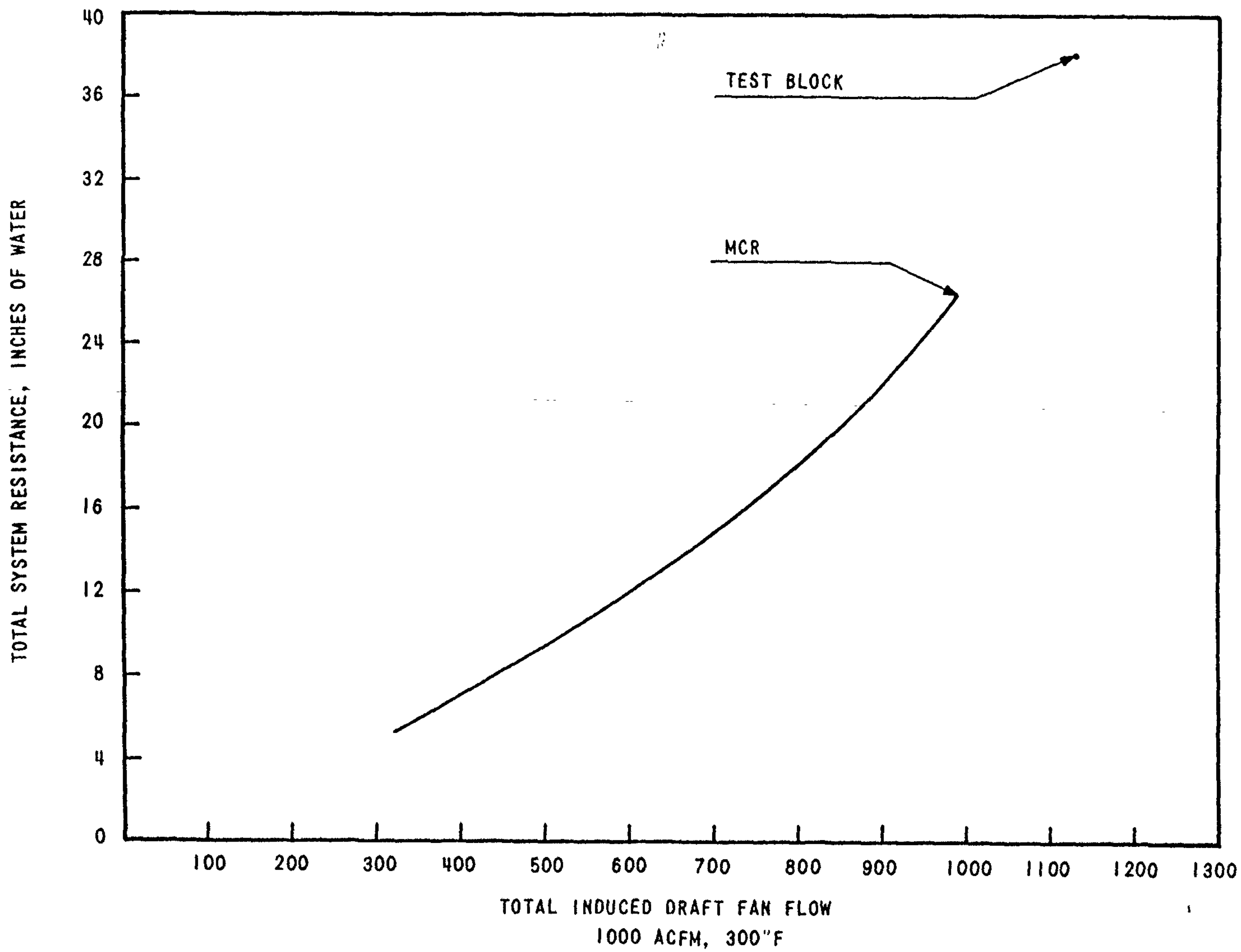
Item	Test Block	MCR 889 MW	100 820 MW	75 615 MW	50 410 MW	25 205 MW
		Generating Unit Load Point				
Inlet Air Temperature, F	300	300	300	300	300	300
Inlet Air Density, lb/ft^3	0.0409	0.0409	0.0409	0.0409	0.0409	0.0409
Capacity, each fan						
Pounds Per Hour	2,769,100	2,436,000	2,291,000	1,852,000	1,363,000	797,000
Actual cfm	1,128,400	992,700	933,600	754,700	555,400	324,800
Fan Static Pressure, in. wg	38.0	26.3	23.8	16.4	11.1	5.3
Fan Static Efficiency, percent	92.18	90.66	90.50	89.89	88.61	83.88
Fan Speed, rpm	954	809	768	636	514	353
Input Horsepower	7,415	4,596	3,918	2,195	1,106	325



INDUCED DRAFT FAN
PERFORMANCE CURVE
FLOW VS STATIC
PRESSURE
FIGURE 3-1



INDUCED DRAFT FAN
PERFORMANCE CURVE
FLOW VS BHP
FIGURE 3-2



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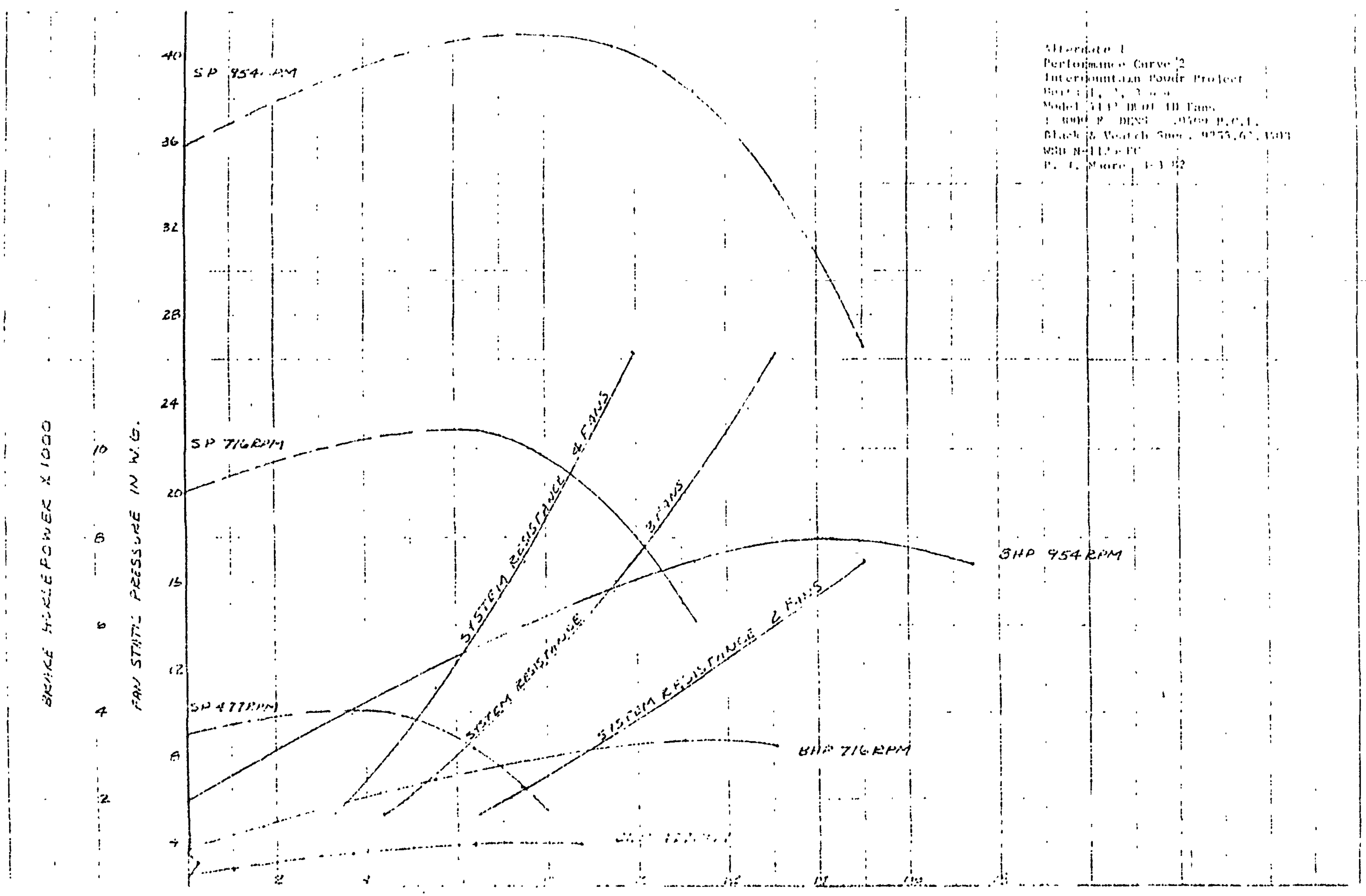
IPP
INTERMOUNTAIN POWER PROJECT

INDUCED DRAFT FAN SYSTEM CURVE

DM-0041



Alternate 1
Performance Curve 2
Intermountain Power Project
Black & Veatch
SP 954 RPM
SP 716RPM
SP 477RPM
BHP 954RPM
BHP 716RPM
SYSTEM RESISTANCE 4 FANS
SYSTEM RESISTANCE 3 FANS
SYSTEM RESISTANCE 2 FANS
FAN STATIC PRESSURE IN W.G.
BRAKE HORSEPOWER X 1000
40
36
32
28
24
20
16
12
8
4
10
8
6
4
2

IP7_039016

TABLE 3-1. FORCED DRAFT FAN PREDICTED PERFORMANCE

TLT Variable Pitch Axial fan (16 blade) [2]
2speed 880/700rpm (4305/2018 HP)

Item	Load Point: Test Block	MCR	100	75	50	25
Inlet Air Temperature, F	110	110	110	110	110	110
Inlet Air Density, lb/ft^3	0.0580	0.0580	0.0580	0.0580	0.0580	0.0580
Capacity, each fan						
Pounds per hour	4,018,300	3,335,600	3,135,800	2,461,800	1,712,200	904,100
Actual cfm	1,154,700	958,500	901,100	707,400	492,000	259,800
Fan Static Pressure, in. wg	19.5	10.9	10.0	7.2	4.8	3.0
Fan Static Efficiency, per cent	84.0	89.7	90.0	90.0	76.5	55.0
Design Fan Speed, rpm	894	717	717	717	717	717
Input Horsepower	4,305	2,018	1,730	966	520	227

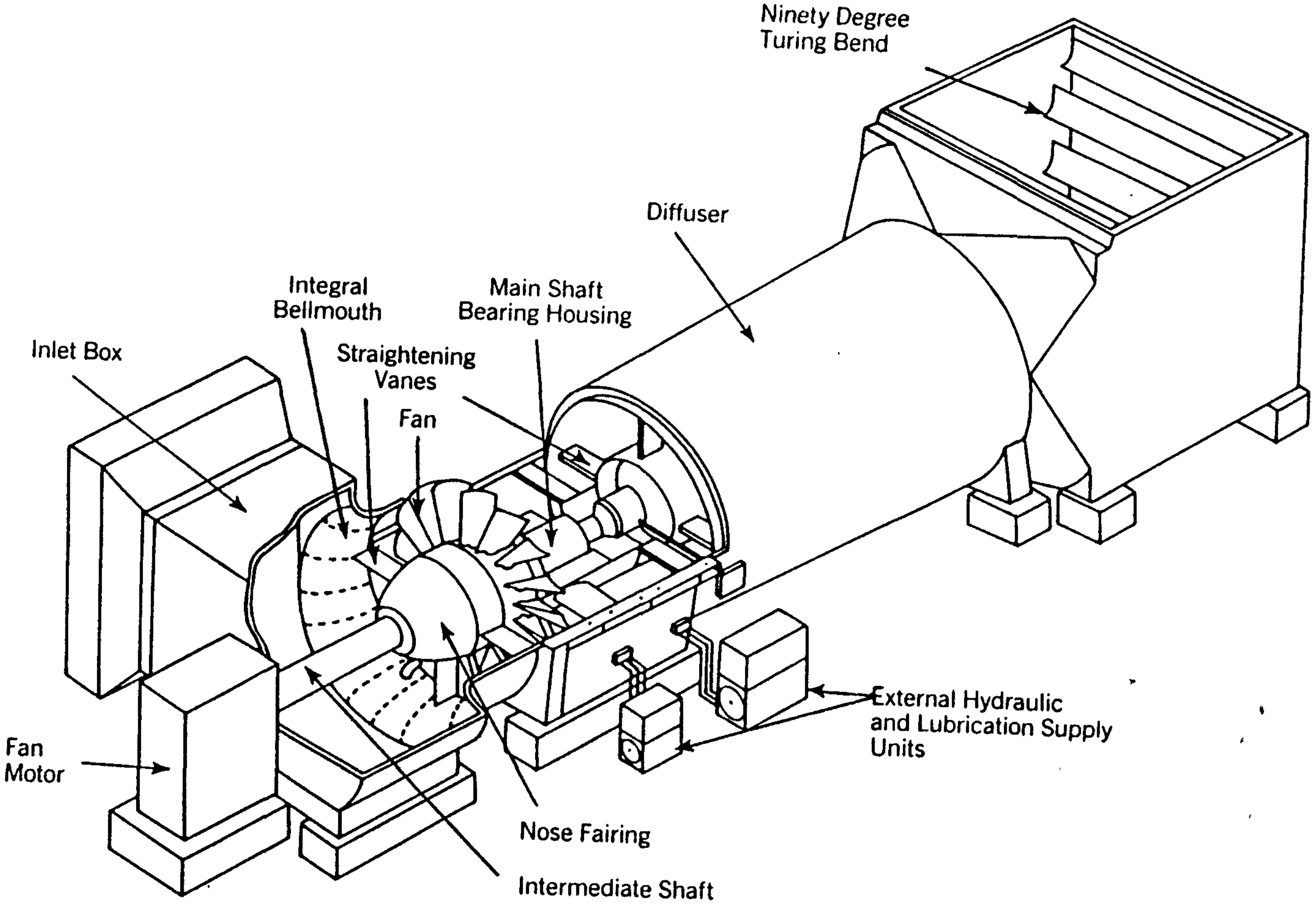
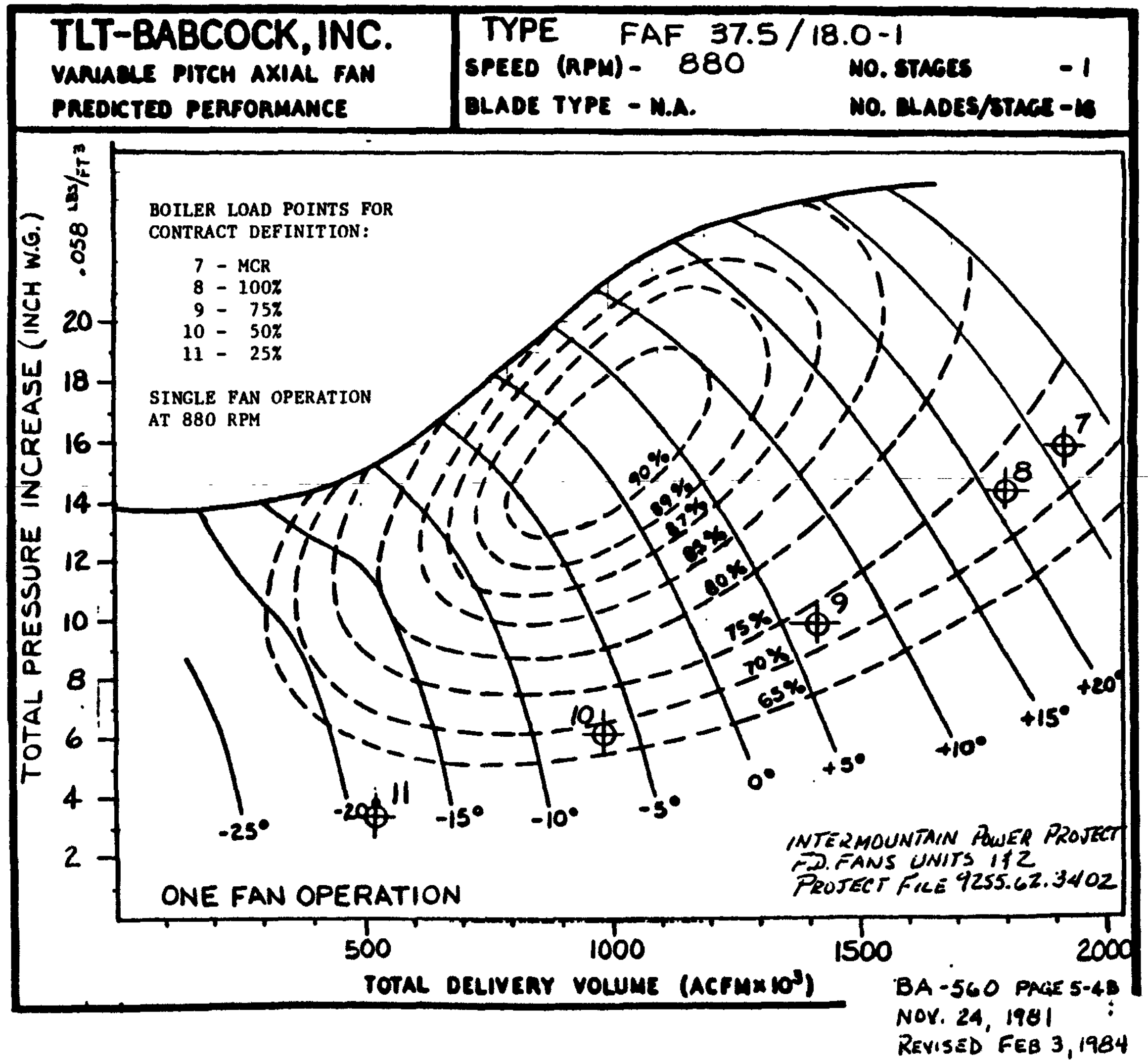
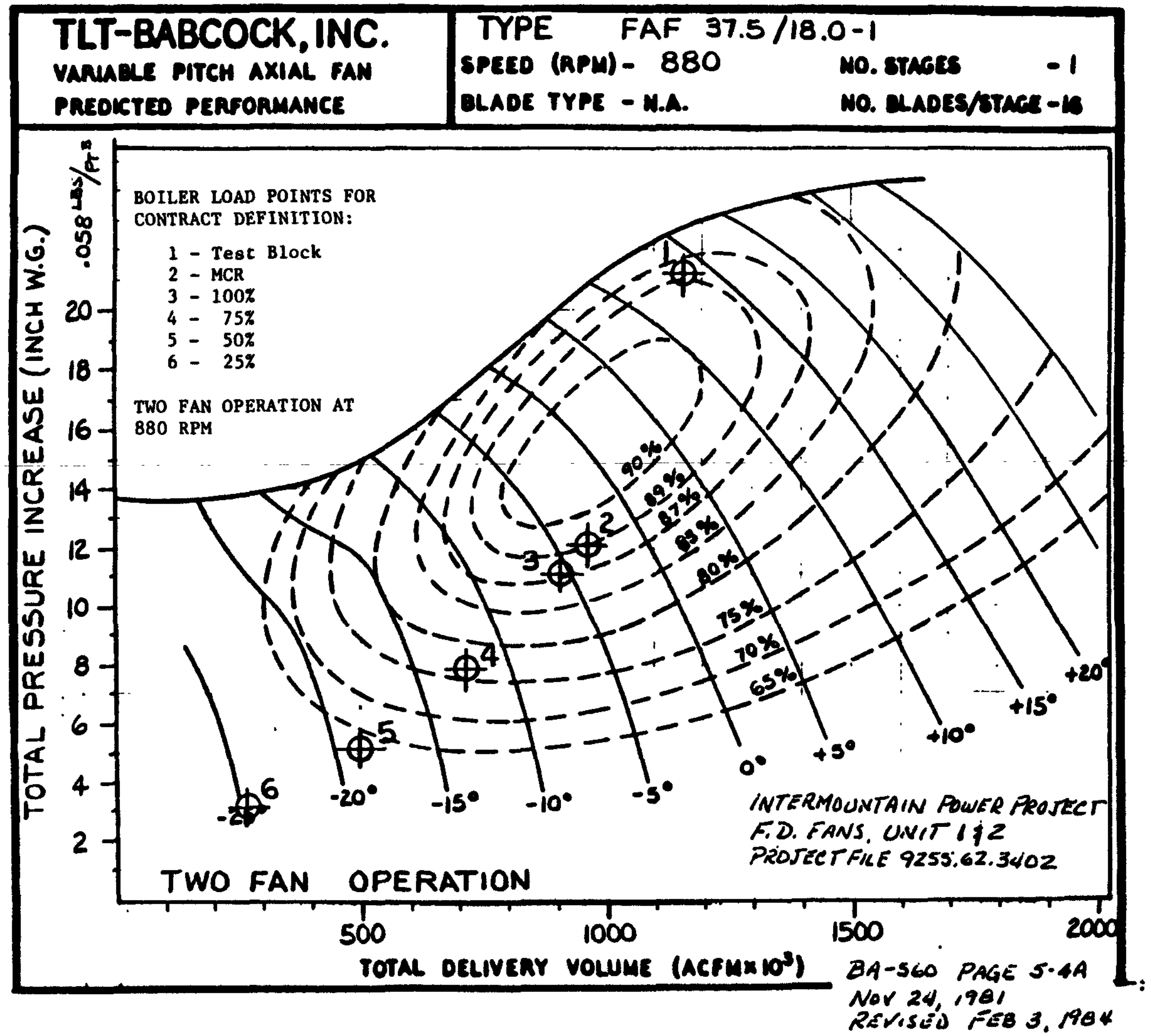


Figure 1

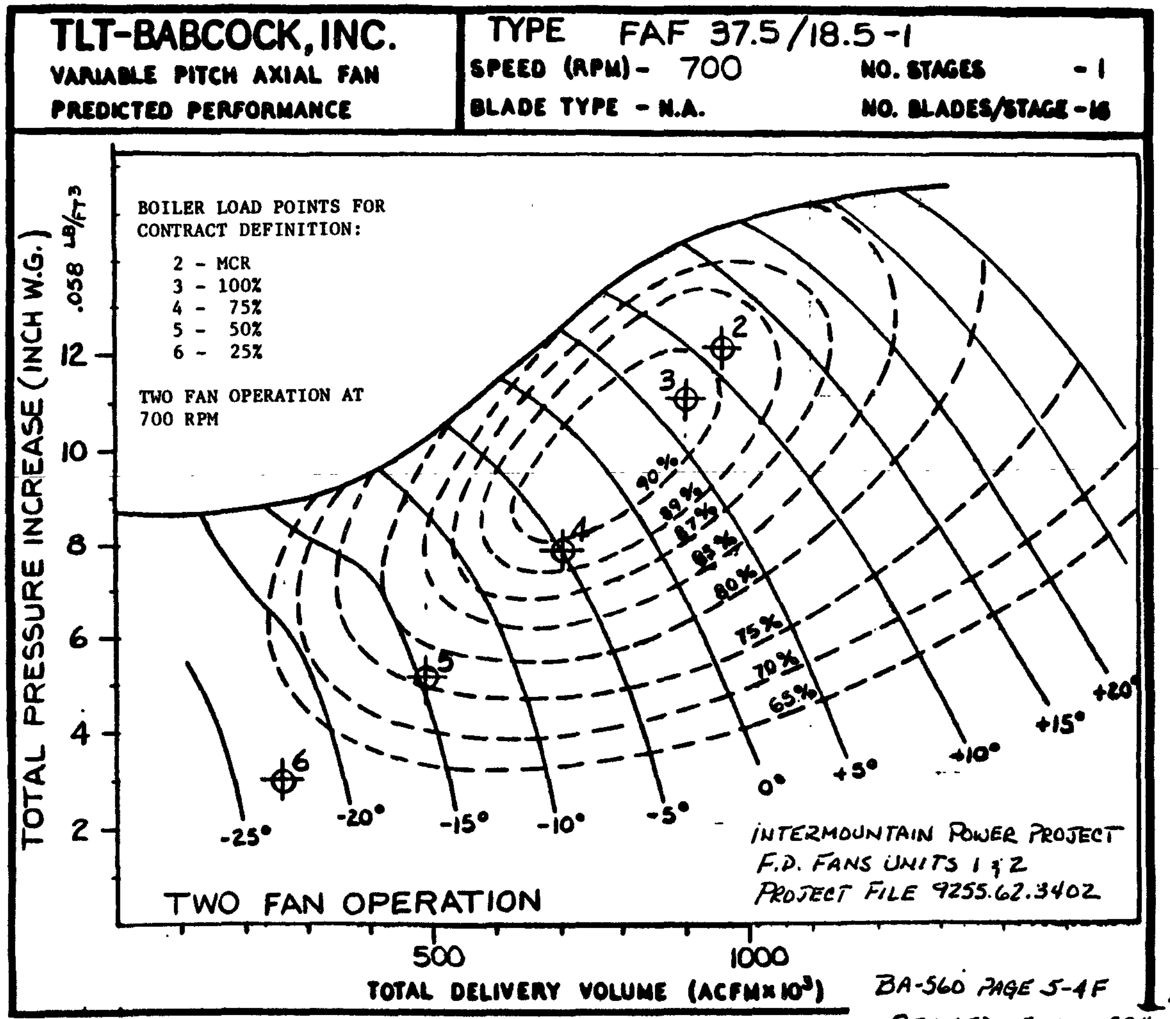
TLT- AXIAL FAN
FORCED DRAFT (2)



FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
880 RPM--SINGLE FAN OPERATION
FIGURE 3-3



FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
880 RPM--TWO FAN OPERATION
FIGURE 3-1



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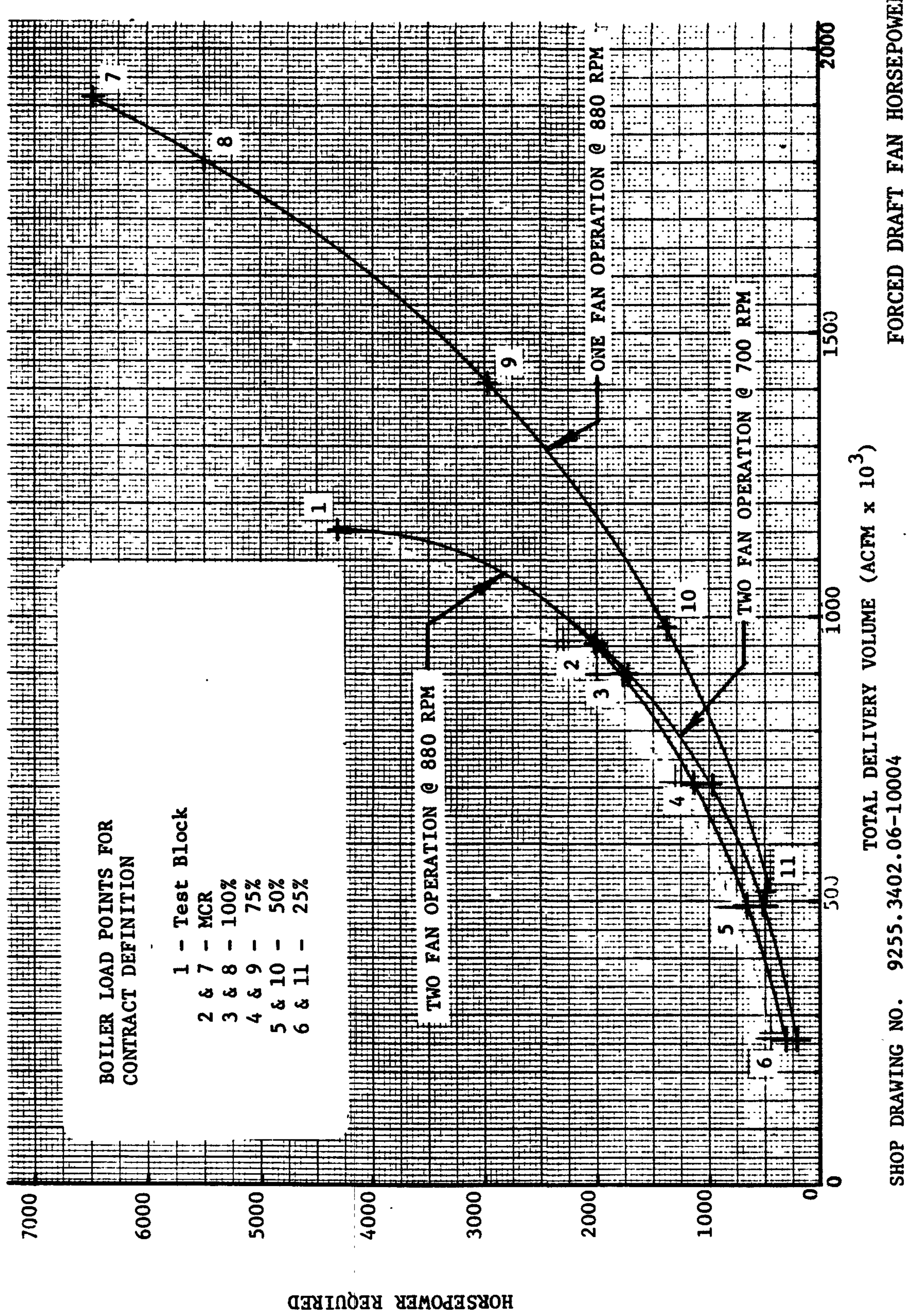
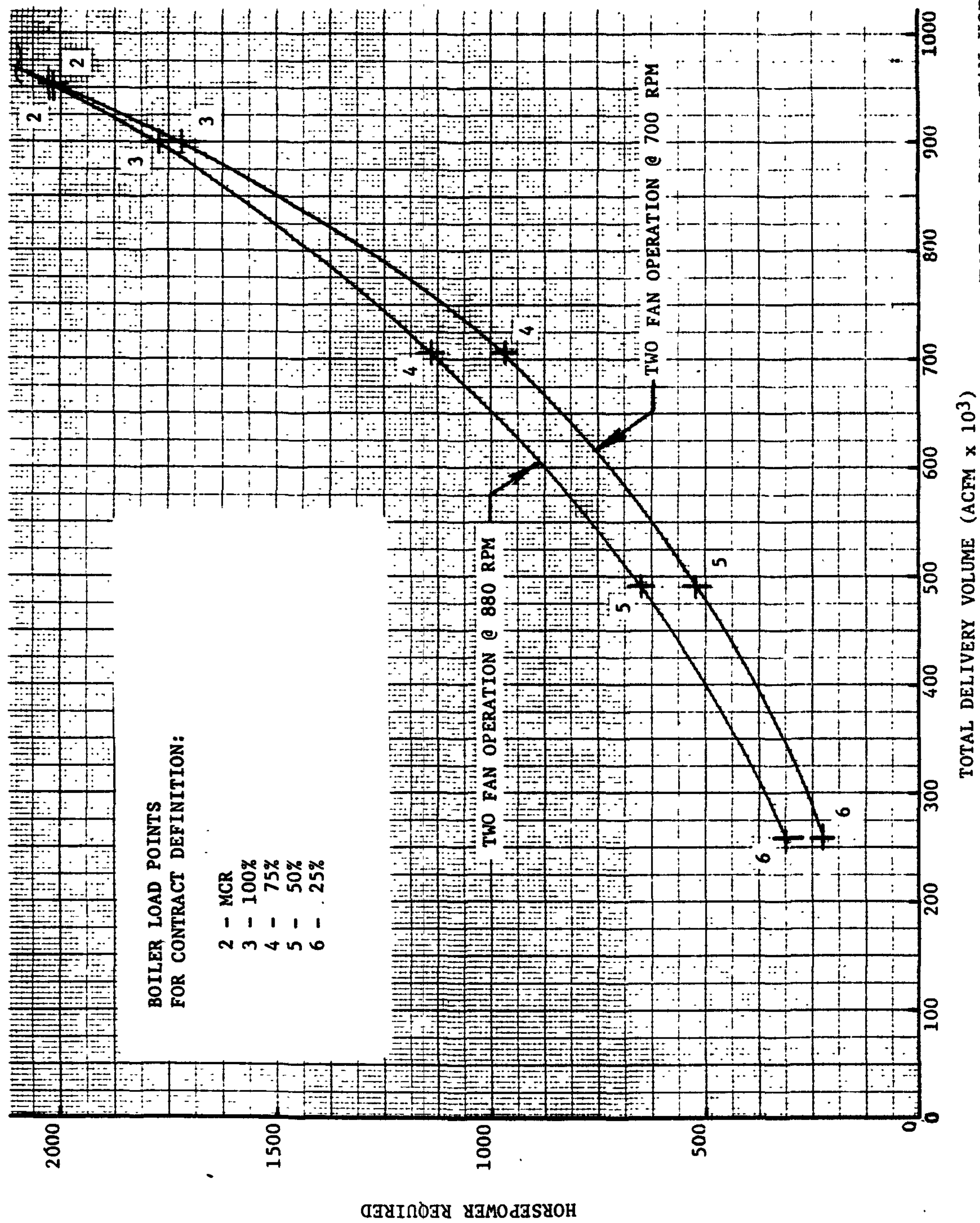
REVISED FEB 3, 1984

FORCED DRAFT FAN STATIC
PRESSURE PERFORMANCE CURVE,
700 RPM--TWO FAN OPERATION
FIGURE 3-2



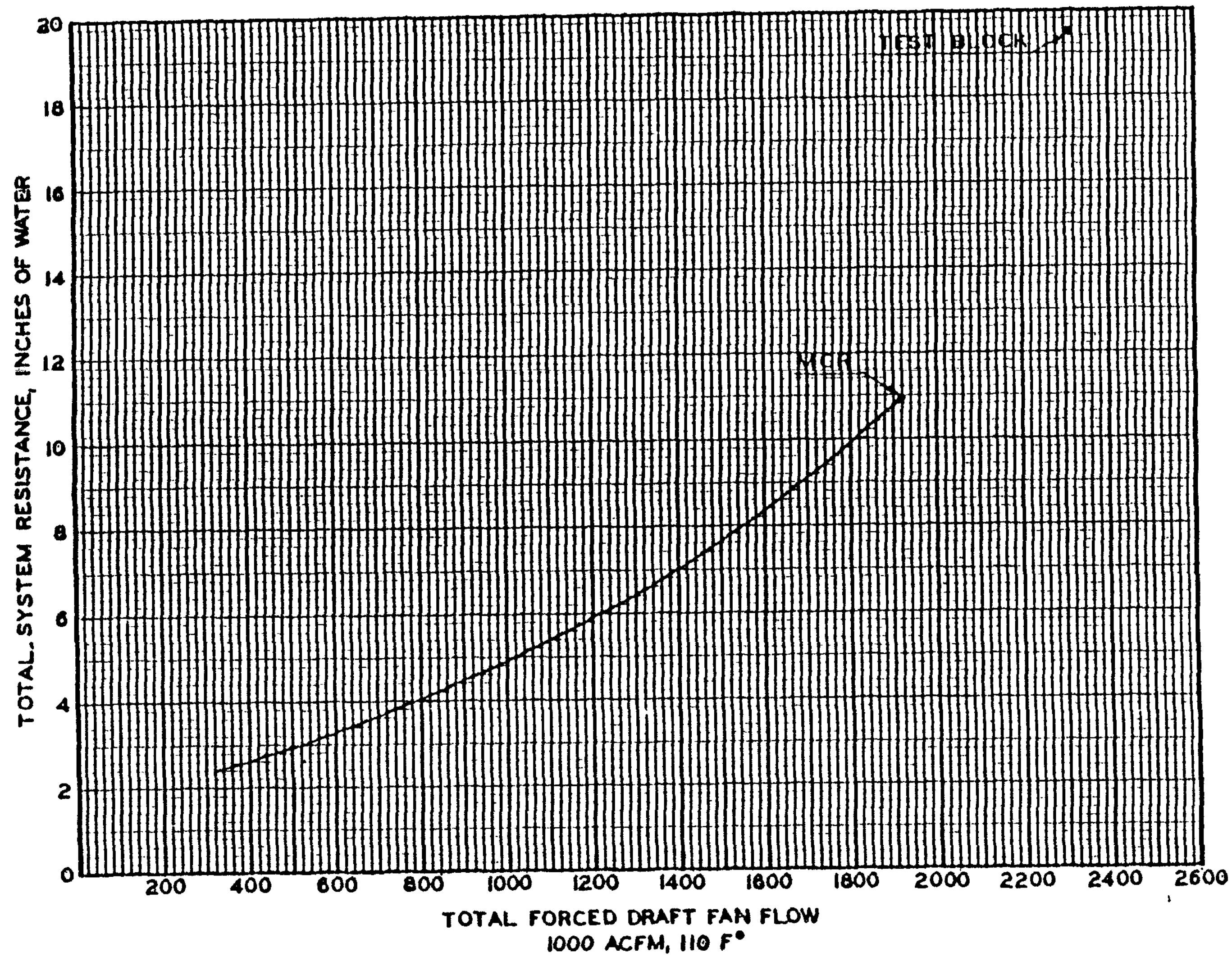
FORCED DRAFT FAN HORSEPOWER PERFORMANCE CURVE

FIGURE 3-4



SHOP DRAWING NO. 9255.3402.06-10006

FORCED DRAFT FAN HORSEPOWER
PERFORMANCE CURVE--
OPERATING LOAD POINTS
FIGURE 3-5



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BLACK & VEATCH CONSULTING ENGINEERS

PROJECT 9255

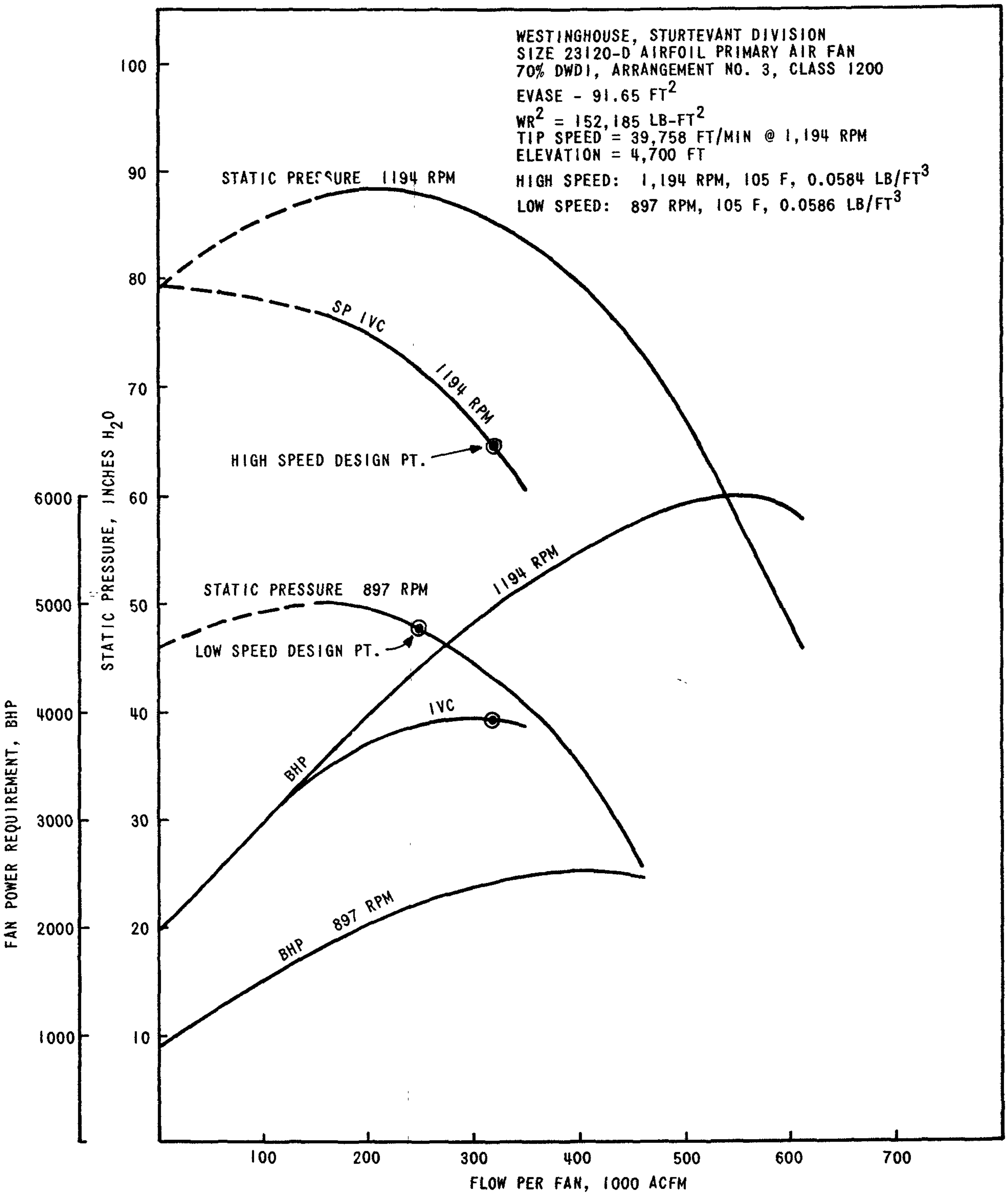
IPP INTERMOUNTAIN POWER PROJECT

FORCED DRAFT FANS SYSTEM CURVE

DM-0032

TABLE 3-2. PRIMARY AIR FAN PREDICTED PERFORMANCE
Westinghouse-Sturtevant Centrifugal airfoil [2]
2 speed (1194/897 rpm) (3810/2061 HP)

Item	Test Block	MCR
Inlet Air Temperature, F	105	105
Inlet Air Density, lb/ft^3	0.0588	0.0588
Capacity, each fan		
Pounds per hour	1,120,300	882,000
Actual cfm	317,500	250,000
Fan Static Pressure, in. wg	62.5	44.5
Fan Static Efficiency, per cent	81.9	84.9
Design Fan Speed, rpm	1,194	897
Input Horsepower	3,810	2,061

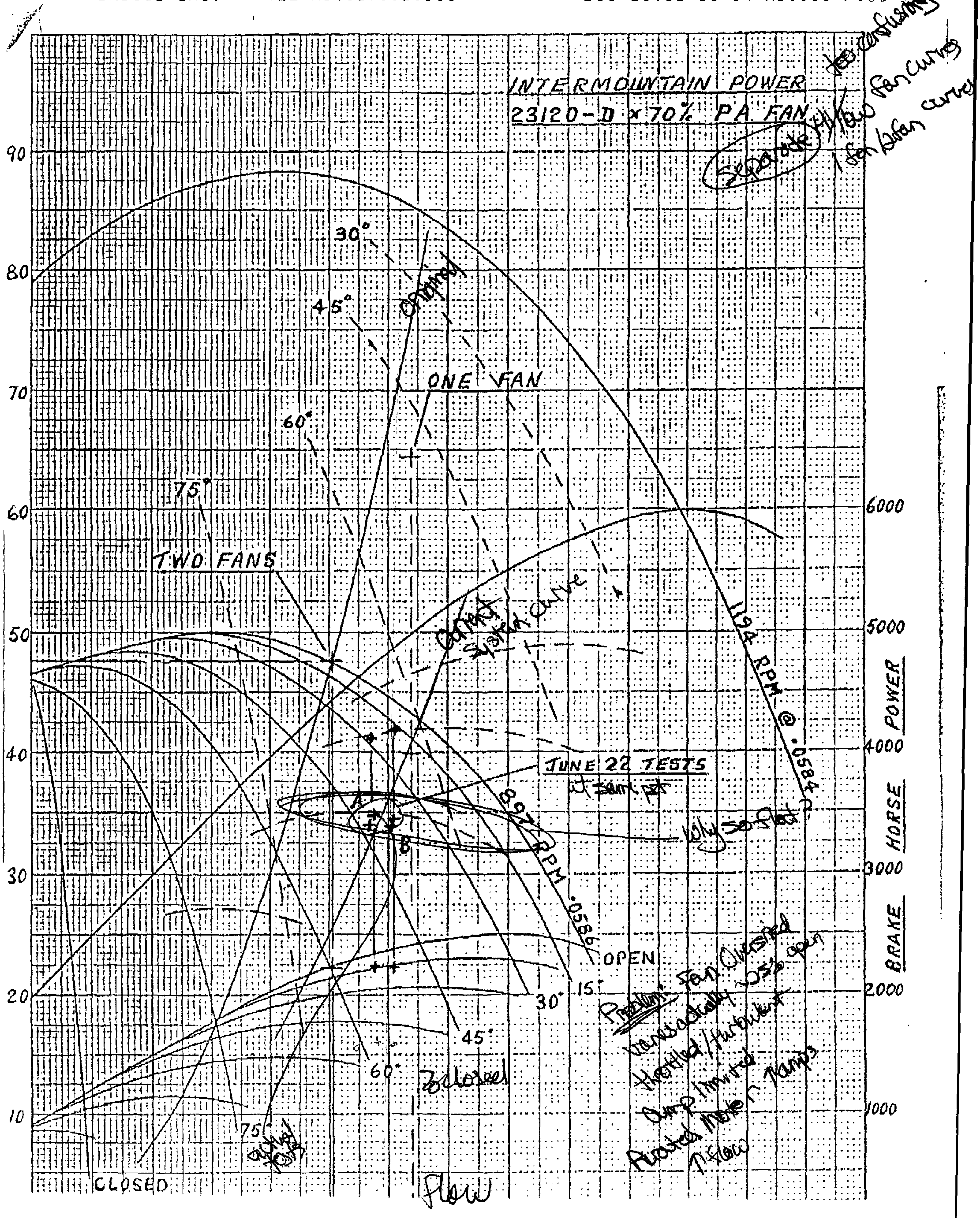


PRIMARY AIR FAN STATIC
PRESSURE AND HORSEPOWER
PERFORMANCE CURVES
FIGURE 3-6

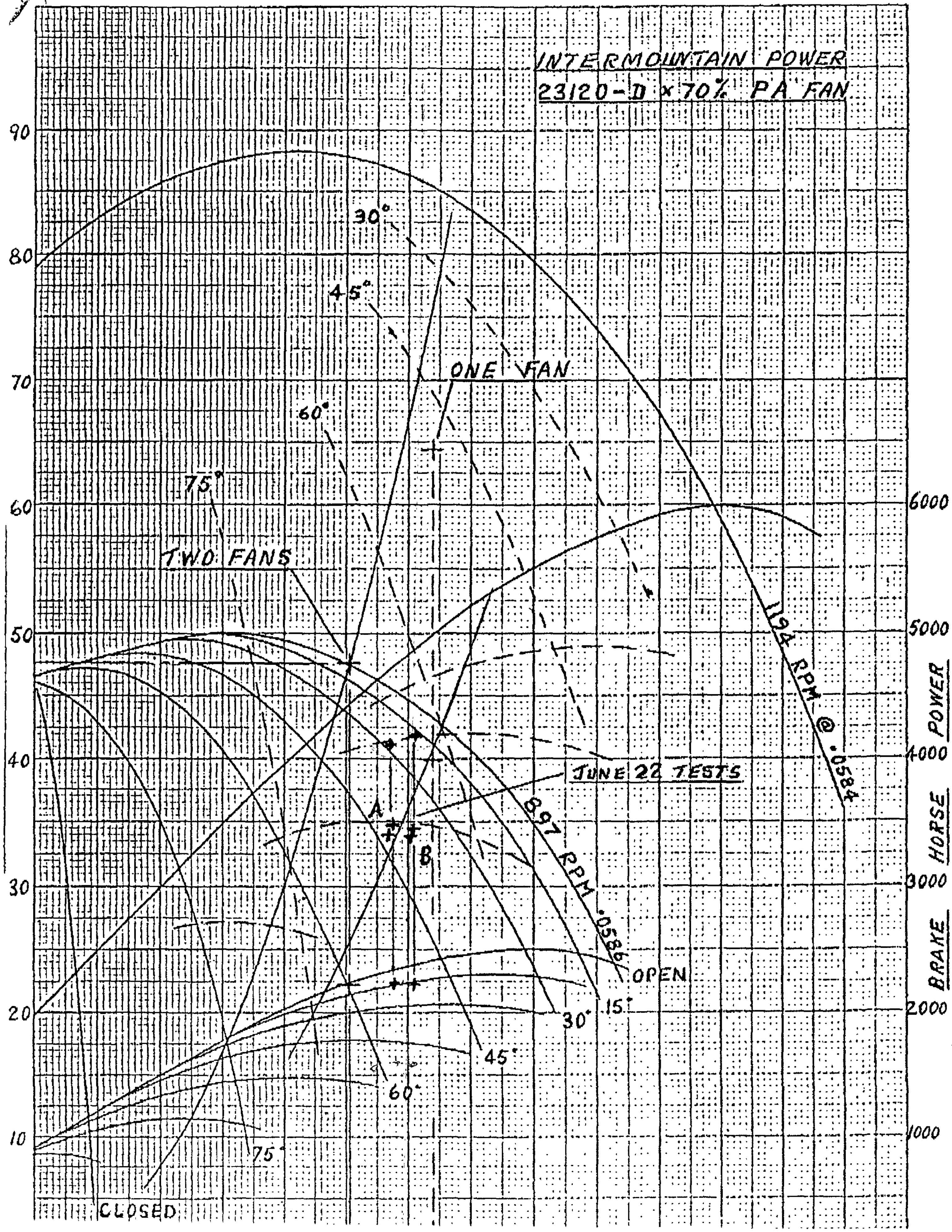
IP7_039026

PA Fan

HOWDEN SIROCCO INC. TEL No.6173613506 Dec 10,91 16:04 No.003 P.09



IP7_039027

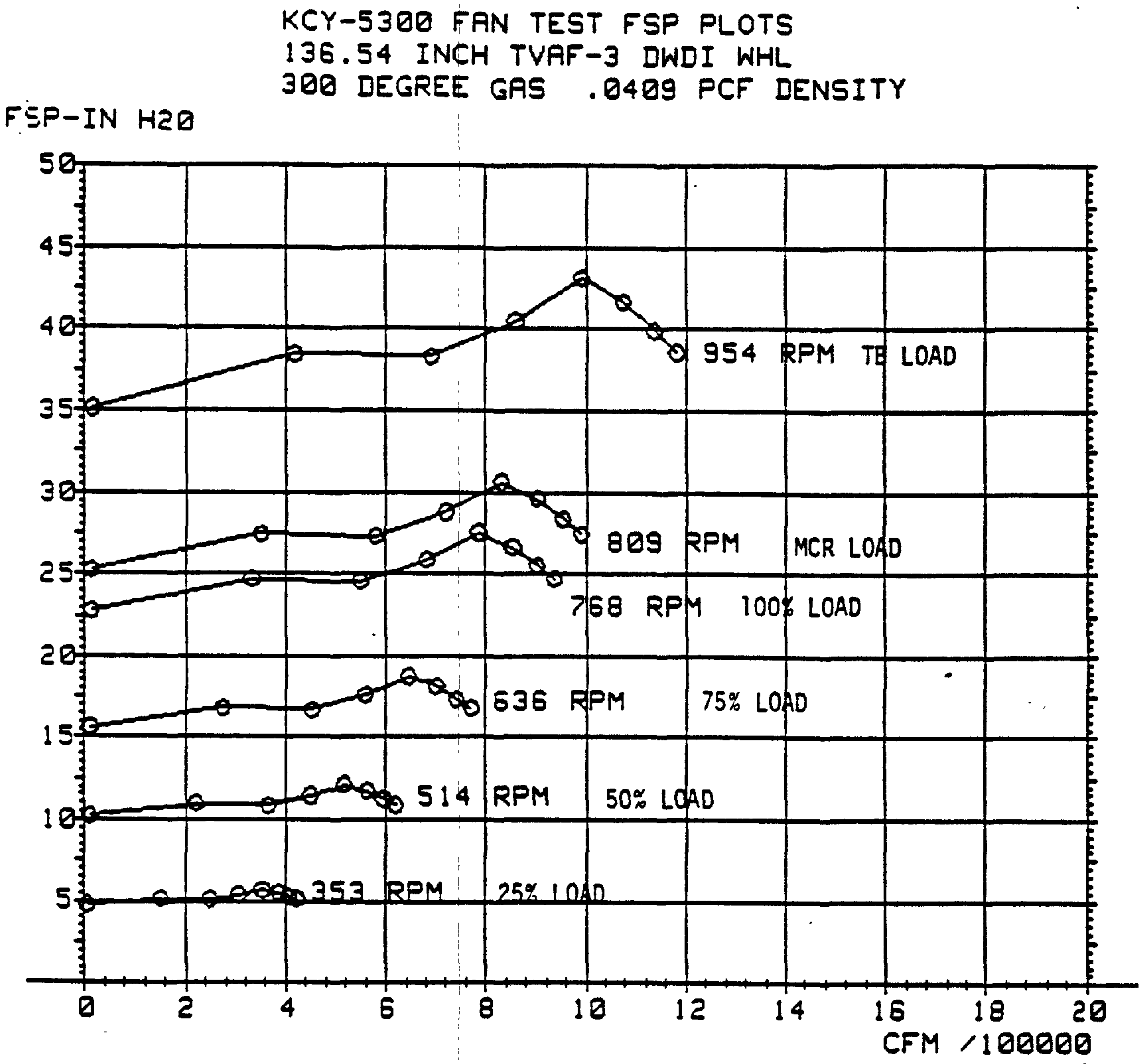

INTERMOUNTAIN POWER
23120-D x 70% PA FAN
90
80
70
60
50
40
30
20
10
6000
5000
4000
3000
2000
1000
BRAKE HORSE POWER
30°
45°
60°
75°
ONE FAN
TWO FANS
JUNE 22 TESTS
A
B
1194 RPM @ .0584
897 RPM .0586
OPEN
15°
30°
45°
60°
75°
CLOSED

ID fans - centrifugal (airfoil, double width, double inlet) 4 ~ 25%
adjustable speed brushless synchronous motor - 7415 HP (W)

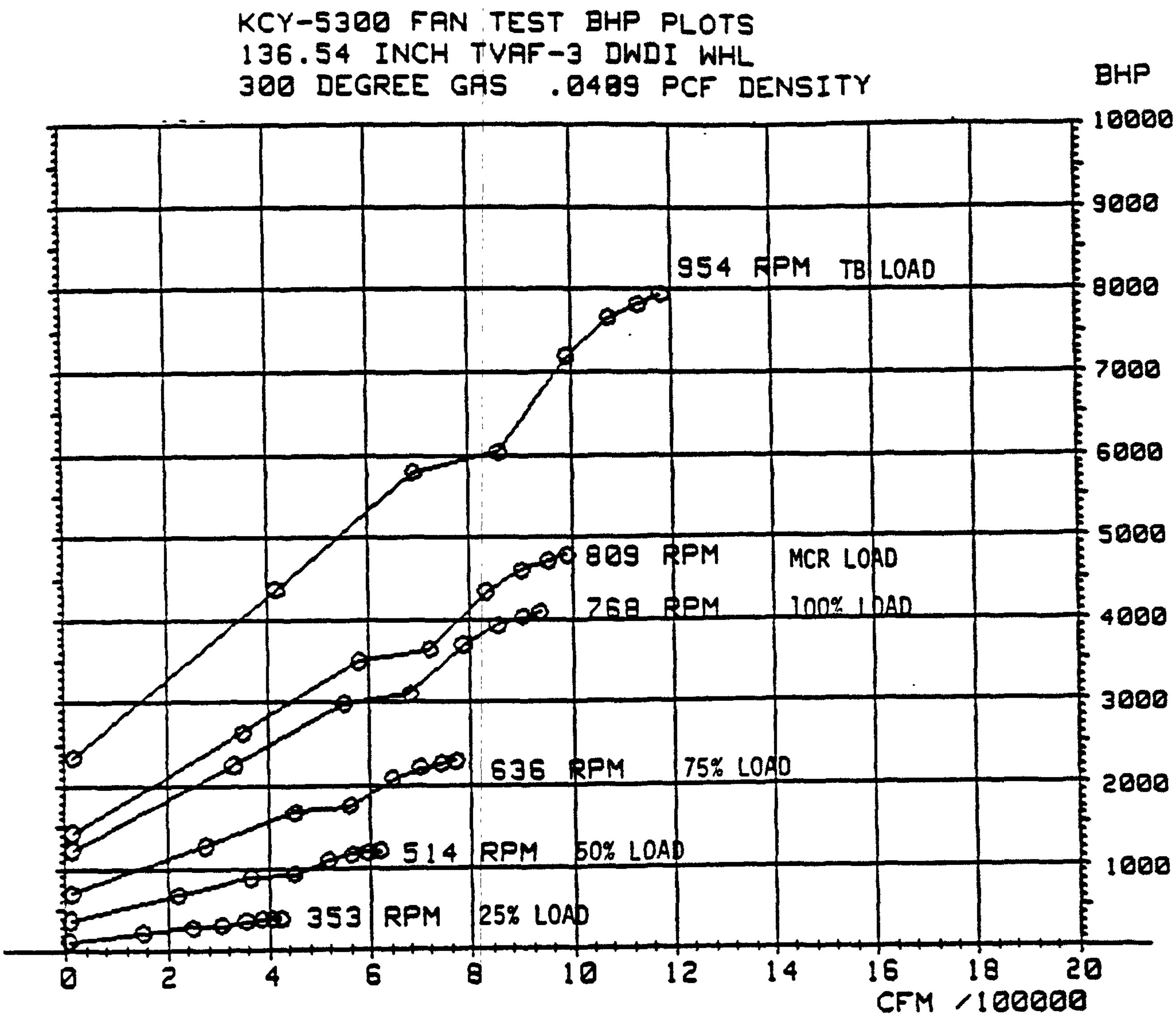
TABLE 3-1. INDUCED DRAFT FAN DESIGN CONDITIONS

Westinghouse - Sturtevant

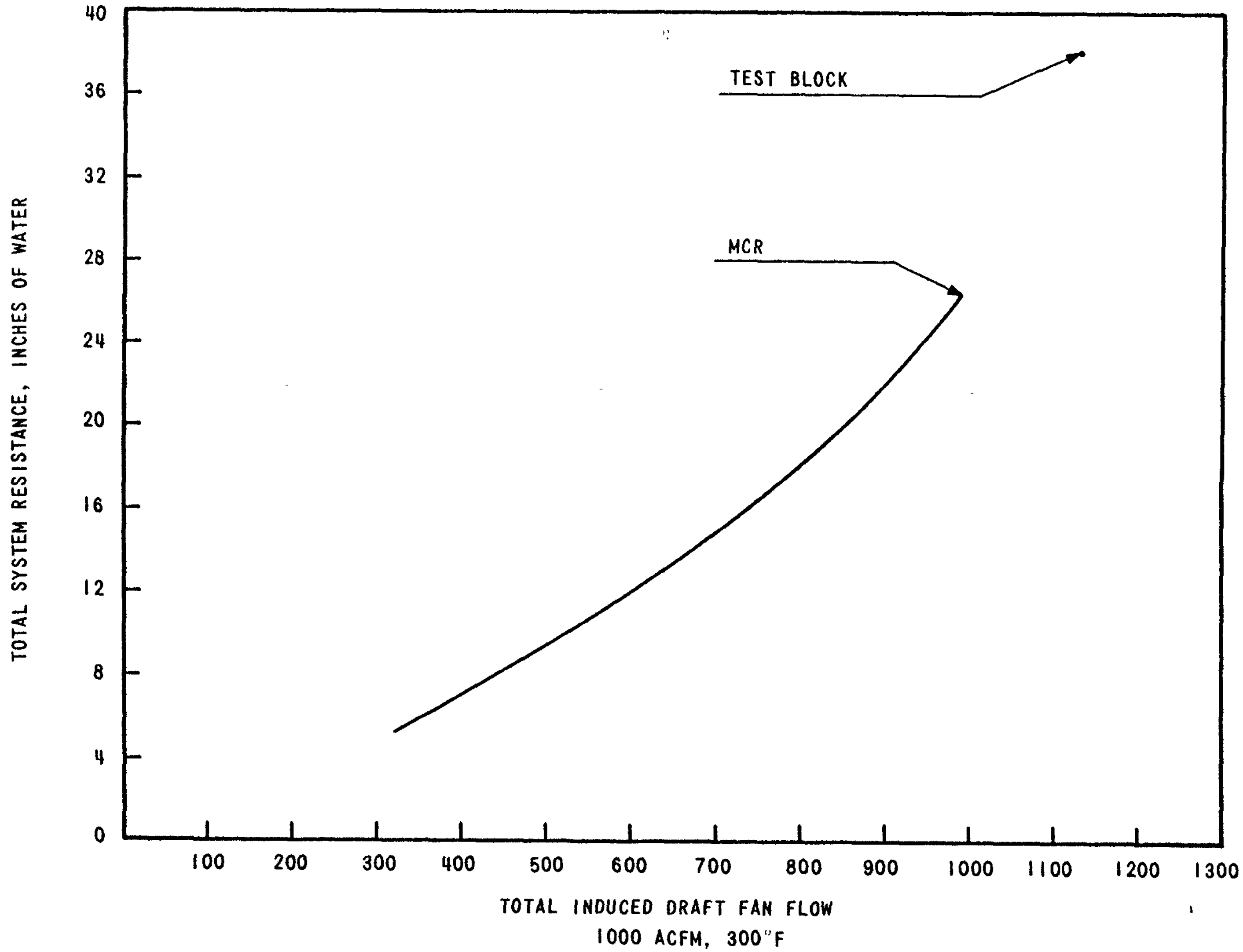
Item	Test Block	Generating Unit Load Point MCR 889 MW	100 820 MW	75 615 MW	50 410 MW	25 205 MW
Inlet Air Temperature, F	300	300	300	300	300	300
Inlet Air Density, lb/ft^3	0.0409	0.0409	0.0409	0.0409	0.0409	0.0409
Capacity, each fan						
Pounds Per Hour	2,769,100	2,436,000	2,291,000	1,852,000	1,363,000	797,000
Actual cfm	1,128,400	992,700	933,600	754,700	555,400	324,800
Fan Static Pressure, in. wg	38.0	26.3	23.8	16.4	11.1	5.3
Fan Static Efficiency, percent	92.18	90.66	90.50	89.89	88.61	83.88
Fan Speed, rpm	954	809	768	636	514	353
Input Horsepower	7,415	4,596	3,918	2,195	1,106	325



INDUCED DRAFT FAN
PERFORMANCE CURVE
FLOW VS STATIC
PRESSURE
FIGURE 3-1



INDUCED DRAFT FAN
PERFORMANCE CURVE
FLOW VS BHP
FIGURE 3-2



NO.	DATE	REVISION	DWN	CK	ACC	APP

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PROJECT
9255

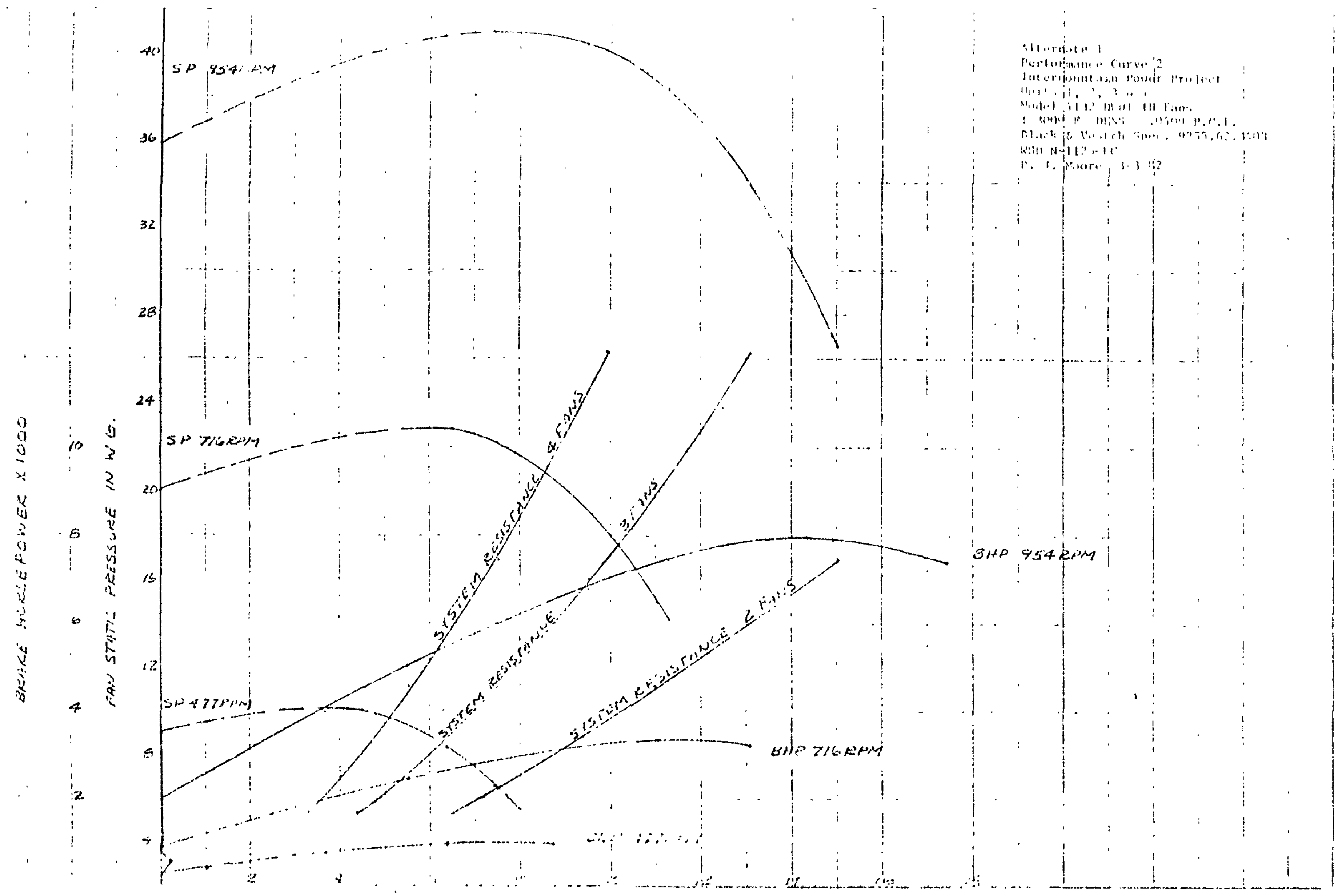
IPP INTERMOUNTAIN POWER PROJECT

INDUCED DRAFT FAN SYSTEM CURVE

DM-0041

2A-8

IP7_039032


Alternate 1
Performance Curve 2
Intermountain Power Project
SP 954 RPM
SP 716 RPM
SP 477 RPM
BHP 954 RPM
BHP 716 RPM
SYSTEM RESISTANCE 4 FANS
SYSTEM RESISTANCE 3 FANS
SYSTEM RESISTANCE 2 FANS
FAN STATIC PRESSURE IN W.G.
BRAKE HORSEPOWER X 1000
40
36
32
28
24
20
16
12
8
4
10
8
6
4
2

6.0 SYSTEM OPERATION

The System Operating Procedure must be reviewed before operating the system.

The FD fans and ID fans are interlocked such that at least one ID fan must be running before an FD fan can be started. Normal starting sequence is one ID fan, one FD fan and then a second ID fan. Refer to the Induced Draft System (CCE) for the ID fan start procedure.

Before starting an FD fan, the blade pitch must be set at minimum and the discharge damper must be closed. For the first FD fan started this will occur automatically after the FD fan control switch is placed in the START position. A gas path must be established by using the same damper alignment used in starting the ID fan. A "gas path integrity" signal is derived by establishing a flow path through an FD fan, an ID fan, a secondary air heater (air and gas side), and any two scrubbers. For specific dampers, refer to the Unit Protection System Description (File No. 9255.93.2403). The local selector switches for the associated FD fan lube oil pump must be set properly; one oil pump set in RUN, while the other oil pump set in STANDBY.

When the FD fan control switch is placed in the START position, the FD fan will start provided all other permissives are satisfied.

The FD fan blade pitch is controlled automatically by the Coordinated Control System when the Hand/Auto control station is placed in the AUTO position. The CCS will control the FD fan blade pitch to maintain airflow.

A PA fan may be started after both UTR's have been reset and in a similar manner as the FD fan. Before starting a PA fan, the associated discharge damper and the inlet vanes must be closed. For the first PA fan started, the discharge damper will close automatically after the PA fan control switch is placed in START position. The local selector switches for the associated PA fan lube oil pumps must be set properly; one oil pump placed in RUN, while the other is placed in STANDBY.

When the PA fan control switch is placed in the START position, it will start provided all other permissives are satisfied. After a time delay to allow the PA fan motor to reach operating speed, the associated PA fan discharge damper will open. The operator may control the PA duct pressure by manually adjusting the inlet vanes, through the software Hand/Auto duct pressure control station or through the hardware Hand/Computer duct pressure control stations. The duct pressure may be automatically controlled by the Coordinated Control System (COA) by transferring all control stations to AUTO.

Due to the increase of motor winding temperature during a fan start, the number of successive starts of each FD and PA fan motor is limited. The number of successive starts of each fan motor should be in accordance with the following.

(1) Motor cold--Two consecutive starts.

(2) Motor at operating temperature--One consecutive start.

Starts subsequent to the above should be 20 minutes apart (with motor running between starts) or 45 minutes apart (with motor not running between starts).

A fan can be stopped by momentarily operating the control switch to STOP. The associated oil pump should be kept in operation to maintain proper lube oil system temperature for a minimum of 10 minutes after an FD fan impeller has come to a standstill. For a PA fan, the oil pump may be stopped after the impeller has come to a standstill.

Typically, a PA and FD fan should be operated on low speed. The high speed mode is utilized only if one fan is out of service or if excessive draft losses are incurred such as when the air heater has become excessively dirty.

7.0 REFERENCE MATERIALS

(1) System Design Specification for Combustion Air (SGB)--9255.43.5802.

(2) System Operating Procedure for Combustion Air (SGB)--9255.95.5802.

(3) Pipeline List--Technical Information; Information Management Report M-PLN-T1 (listed by System Code for Unit 1, Unit 2, and Common).

(4) Valve List--Technical Information; Information Management Report M-V-T1 (listed by System Code for Unit 1, Unit 2, and Common).

(5) Manufacturer Drawings are listed by contract number in the following reports.

(a) Manufacturer Drawings--Status and Distribution Report; Information Management Report PJ-DM-DST-L1.

(b) Manufacturer Drawings--Cross Reference Report; Information Management Report PJ-DM-CRF-L1.

Purchase and installation contract numbers are listed by System Code for Unit 1, Unit 2, and Common in "Major Equipment--General Information Management Report G-EQ-G1A."

(6) Plant Arrangement Drawings.

(a) 9255-1BSA-M1028A through -M1032.

(b) 9255-2BSA-M1028A through -M1032.

(c) 9255-9BSA-M1001.

(7) Piping and Instrument Diagrams.

(a) 9255-1SGB-M2064A through -M2064C.

(b) 9255-2SGB-M2064A through -M2064C.

(8) Logic Diagrams--9255-1SGB-K1000.

(9) Detail Piping Drawings.

(a) 9255-1SGB-M4089A.

(b) 9255-1SGB-M4089B.

(c) 9255-1SGB-M4090.

(d) 9255-2SGB-M4089A.

(e) 9255-2SGB-M4089B.

(f) 9255-2SGB-M4090.

(10) Schematic Diagrams--K2000 Series; listed by System Code for Unit 1, Unit 2, and Common in "B&V Drawings--Drawing Status Report; Information Management Report PJ-DB-ST-L2."

(11) Instrument Connection Schematics--K8000 Series; listed by System Code for Unit 1, Unit 2, and Common in "B&V Drawings--Drawing Status Report; Information Management Report PJ-DB-ST-L2."

(12) Purchase Documents.

(a) Forced Draft Fans--9255.62.3402.

TLT-Babcock, Inc.

- Forced Draft Fans.
  Instruction Book for Axial Fans.
- Witness Test Report--April 1984, Report No. RAS 84-007.
- Forced Draft Fan Hydraulic Control Units.
  Akro-Tec, Inc.
  Instruction Book for Control Units.
- Forced Draft Fan Lube Oil Units.
  Akro-Tec, Inc.
  Instruction Book for Lube Oil Units.

(b) Electric Motors--9255.63.2201.1.

Westinghouse Electric Corporation.

- Forced Draft Fan Drive Motors.
  Instruction Book for 6,900 Volt Motors.
- Forced Draft Fan Speed Change Switches.
  Esco Manufacturing Co.
  Instruction Book for Speed Change Switches.
- Primary Air Fan Drive Motors.
  Instruction Book for 6,900 Volt Motors.
- Primary Air Fan Speed Change Switches.
  Esco Manufacturing Co.
  Instruction Book for Speed Change Switches.

(c) Steam Generator--9255.62.3401.

Babcock & Wilcox.

- Primary Air Fans.

  Westinghouse Electric Corporation

  Instruction Book for Centrifugal Fans.

- Primary Air Fan Lube Oil Units.

  Akro-Tec, Inc.

  Instruction Book for Lube Oil Units.

- Secondary Air Heaters.

  C-E Air Preheater Company

  Instruction Book for Air Heaters.

- Primary Air Heaters.

  C-E Air Preheater Company

  Instruction Book for Air Heaters.

- Air Heater Bearing Lube Oil Units.

  C-E Air Preheater Company

  Instruction Book for Air Heaters.

Section 1.2 Contract Data Sheets

WSD General Order No. AKY-6081 Item No. 1&2 (Combined = Serial No.)

Contract/Main Assembly Drawing No. 2090F60

Purchaser: Babcock & Wilcox Company, #334-0614/P.O. 336601DU

User: Intermountain Power Agency

Intermountain Power Project

Plant or Station: Intermountain Generating Station ; Unit No. 1

User Equipment Identification No. IPA Contract 2010N

Consulting Engineer: Black & Veach

Job No. 9255.62.3401 ; Specification No. 2010N

1. FAN DESCRIPTION: Series/Size 23120 : D Width D Inlet
   Speed Class 1200/900 RPM; Wheel Type Airfoil

2. APPLICATION:

DUTY - Forced Draft ☐
Induced Draft ☐ FUEL ______
Primary Air ☒
Hot Gas Recirculation ☐ FUEL ______
Other ______

DUTY CYCLE - Continuous Operation ☒
Cyclical Operation:
Shut Down/Start-Up Weekly ☐
Shut Down/Start-Up Daily ☐

TEMPERATURE:

-20 °F Minimum, 200 °F Maximum - Ambient
80 °F Normal Operating
200 °F Maximum Continuous Operating (Design)
______ °F Maximum Upset for ______ Hours
Not to exceed ______ times per year.

Effective January 1978

Westinghouse Electric Corporation
Sturtevant Division
Hyde Park, Boston, MA 02136

WSD General Order No. AKY-6081 Item No. 1&2

RATE OF TEMPERATURE CHANGE:

3. CONTRACT PERFORMANCE

	FAN LOAD	VOLUME CFM	STATIC PRESSURE INLET	STATIC PRESSURE OUTLET	STATIC PRESSURE RISE	TEMP °F	SPEED RPM	BHP	DENSITY #/FT.^{3}
High Spd.	T.B.	320,000	-3.73	60.71	64.44	105	1194	3989	0.0584
	NET	244,000	-2.40	39.37	41.77	80	1194	3150	0.0614
Low Spd.	T.B.	252,600	-2.36	45.22	47.58	105	897	2217	0.0586
	NET	222,600	-2.16	34.46	36.62	60	897	1739	0.0637

Performance based on elevation of 4700 feet, 25.18" Hg Bar. Pressure

Performance based on use of Evase expanded to INCL sq. ft.

4. NOISE LEVELS

BAND Center Frequency

Measured at

H_Z	1 63	2 125	3 250	4 500	5 1000	6 2000	7 4000	8 8000	Overall
A	75	78	78	72	58	67	77	78	85
B	105	108	115	116	118	118	115	108	124
C	82	85	89	89	90	89	81	69	96
D									

A - Inlet noise, measured 5 feet from inlet
B - Discharge Noise, measured 5 feet from discharge
C - Noise thru housing, measured 5 feet from housing, without inlet noise added
D - Noise around fan, measured ____ feet from housing with inlet noise added, BUT WITHOUT discharge, drive or noise effect of other equipment added.

Above noise levels are:

☐ Sound power Ref. 10^{-12} Watts (metric)
☒ Sound pressure Ref. .0002 microbars
☒ A scale ☐ C scale

January 1978

Revised April 9, 1986

IP7_039040

WSD General Order No. AKY-6081 Item No. 1&2

Comments: ______

5. WHEEL GAUGES/MATERIALS

Hubs: FLANGE 1.50 TK ASTM A441
CYLINDER: STL FORGING SAE 1017-1020-1023

Center/Backplate: 1.25 " Thick: ASTM A514 GR A

Blades:

Single Thickness:

Blades: ______ " Thick: ______

Blade Liner: ______ " Thick: ______

Reinforcing Rings: ______ " Thick: ______

Corner Pads: ______ " Thick: ______

Airfoil:

Skins: .31 " Thick: ASTM A514 GR A

Ribs: .31 " Thick: ASTM A514 GR A

Nose: .62 " DIA Thick: STL BAR SAE 1015

Liner: ______ " Thick: ______

Side Plates: .75 " Thick: ASTM A514 GR A

When Field Balancing of Rotor is necessary follow procedures outlined in Section 16.2 - Field Balancing of Heavy Duty Fan Wheels. Use Category III for welding weights to wheel.

6. BEARINGS: Size 10 " Dia: Type (W) HD SLEEVE TYPE

COOLING MEDIUM:

☐ Water: ______ GPM Max; ______ GPM Min.
Water Temperature: ______ °F Max; ______ °F Min.
Water Pressure: ______ PSI Max; ______ PSI Min.

WSD General Order No. AKY-6081 Item No. 1&2

☐ Air: ______ CFM at ______ "H_2O Static Pressure

Air Temperature: ______ of Max: ______ of Min.

☐ Non-Cooled (Ambient Air)

☒ External Lubricator

LUBRICANT

☐ Grease: Type ______

☒ Oil: Viscosity 300 SSU at 100 °F
(See Section 6 for complete Oil description)

Quantity: ______ Qts. per Bearing - Self Contained

______ GPM Max.: ______ GPM Min. per Bearing - Flood Lubricated

LUBRICATING METHOD

☐ Ring - Self Contained

☐ Disc - Self Contained

☒ Flood Lubricated - From External Lubricator

BEARING OPERATING TEMPERATURE

Babbitt: Per Tip Sensitive Thermocouple or Resistance Temperature Detector

160 °F to 180 °F Normal Operating Range

190 °F Alarm

200 °F Shutdown

Oil temperature for Start-up is 50°F for Self-contained Bearings; 90°F for Flood Lubricated Bearings.

7. LUBRICATOR: Nominal Size (Tank Capacity) ______ Gallons

HEAT EXCHANGER	CIRCUIT
☒ Water-to-Oil	☐ Single
☐ Air-to-Oil	☒ Dual

WSD General Order No. AKY-6081 Item No. 1&2

Electrical Requirements:

Pump Motors: 3 HP, 1800 RPM

3 Phase, 60 HZ, 230/460 Volts

TEFC, 182T

Reservoir Heater: 9.25 KW, 3 Phase ________ 480 Volts

Air Cooled Heat Exchanger Fan Motor: - HP, - RPM,

3 Phase, ________ HZ, ________ Volts

8. TURNING GEAR NONE

Turning Gear Assembly designed to:

☐ Start fan from Rest

☐ Engage during coast down ONLY and maintain specified RPM.

Motor: ________ HP, ________ RPM

3 Phase, ______ H_Z, ________ Volts

Gear: Output RPM ____________ (Input to Fan)

Motor-Gear Designed for:

Fan Rotor: Weight ________ Lbs., WR^2 ________ $Lbs/Ft.^2$

Drive Rotor: Weight ________ Lbs., WR^2 ________ $Lbs/Ft.^2$

Bearing; Dia. ________"; Sleeve Length ________"

WSD General Order No. AKY-6081 Item No. 1&2

9. DRIVE Westinghouse PAM Motor G.O. CL30900
M.A 1391F83

Main Drive: ☒ Motor 4000 HP 1200 RPM
2000HP 900 RPM

3 Phase, 60 H_Z ____ Volts

☐ Single Speed; ☒ Two-Speed; ☐ Variable Speed

☐ Turbine ____ HP ____ RPM at Test Block

____ RPM Maximum Overspeed

Overspeed: Continuous ☐

Momentary ☐

Secondary Drive: ☐ Motor ____ HP, ____ RPM

3 Phase, ____ H_Z ____ Volts

☐ Turbine ____ HP, ____ RPM at Design

____ RPM at Max. Overspeed

Overspeed: Continuous ☐

Momentary ☐

Variable Speed Drive: Type: ____
(Fluid Drive, Magnetic Coupling, etc)

Make: ____

Model: ____ HP ____

RPM - Min. ____ Max. ____

10. VIBRATION AMPLITUDES

The following table indicates the normal and allowable horizontal and vertical vibration levels of our products. Also shown are levels at which corrective action is required. All values shown are in mils; peak-to-peak amplitude as measured on the bearing housing, horizontal centerline with a seismic measuring device.

WSD General Order No. AKY-6081 Item No. 1&2

Operating Speed RPM	Peak-to-Peak Amplitude - Horizontal and Vertical - Mils		
	Normal	Rough Alarm	Correction Required Shutdown
1800	0 to 2.0	3.5	5.0
1500	0 to 2.0	4.0	5.5
1200	0 to 2.5	4.5	7.0
1000	0 to 3.0	5.5	7.5
900	0 to 3.0	6.0	8.5
750	0 to 3.5	7.0	9.5
720	0 to 3.5	7.0	9.5
600	0 to 4.5	8.0	11.0
514	0 to 5.0	8.5	12.5

NOTE: For allowable Axial vibration levels, use one-half of above values.

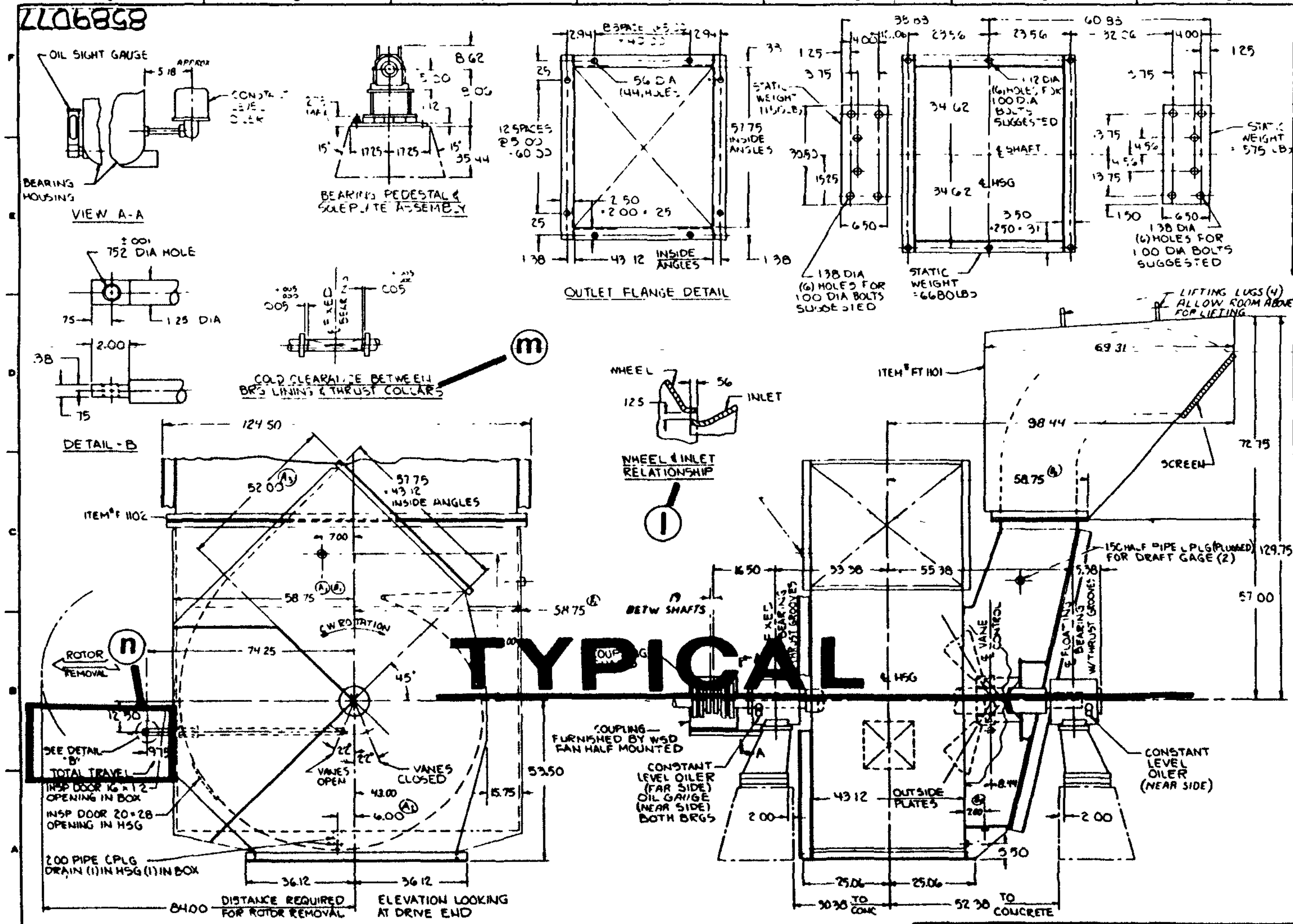
11. SUPPLEMENTAL DRAWINGS

DRAWING NO.	DESCRIPTION	SECTION
2426D12	FOUNDATION SYSTEM DESIGN CRITERIA	4
2090F60	~~CONTRACT DWG 23120~~	5
7645A45	HARDWARE	5
7645A58	TEMP DETECTOR (BOX)	5
5453C87	~~BRACING ROD ASSY~~	5
7646A64	TEMP DETECTOR (BRGS)	6

TITLE CONTRACT DWG 4054-B

DWG 8589D77 REV 1,2,3,4,5

QTY	NOTE	MATL SIZE CODE PART NUMBER	ITEM	COMPONENT DESCRIPTION	REFERENCE INFORMATION
1		8592D16G001	1	HOUSING & BOX ASSY	PC
1		5243C12G001	2	ROTOR ASSY	(55.38 EFF DIA)
1		6877A64G004	3	BRG FIXED	2.94 SAWC/SEALS
1		6877A64G005	4	BRG FLTG	2.94 SAWC/SEALS
2		1596B77G001	5	PEDESTAL & SOLEPLATE ASSY	
1		7624A20G003	6	VANE (INLET	ASSY 8A (CCW)
2		9C1154H047	7	SEAL RING	
2		9C1154H046	8	HOLDING RING	
1		8592D28G001	9	VANE LINKAGE	
1		8592D23G001	10	ELBOW (SOUND TREATED)	
1		7626A71G001	11	HARDWARE LIST	
2	D	3326B21G001	12	LEVEL OILER	
1	K	8589D77H013	13	COUPLING	FAN/MOTOR
1		3951B56G001	14	COUPLING	GUARD



A- HOUSING & BOX .25 TK STL BRACED AS REQUIRED.
B- SERVICE SPLITS SHOWN ———— ALL BRACING SPLITS TO BE WELDED AT FIELD ASSY BY CUSTOMER. INLET & VANE ASSY TO BE ASSEMBLED IN FIELD PER DRAWING 2299C06.
C- VANE CONTROL TORQUE = 130 FT/LBS
D- BEARING TO HAVE TRICO CONSTANT LEVEL OILER 8 OZ.
E- FAN DESIGNED FOR MAX TEMP OF 150°F
F- ROTOR WGT = 900 LBS. ROTOR WR^2 = 1720 LBS FT^2
G- THE STATIC VERTICAL WEIGHT LOAD OF THE SOUND ELBOW IS SUPPORTED BY FAN HOUSING FOR ANY OTHER LOADING (WIND, ETC) CUSTOMER MUST PROVIDE ADDITIONAL TIES AND SUPPORTS
H- FOR SUPPLEMENTAL DWG LIST SEE DWG 7626A78
J- PAINT ALL PARTS WITH ONE COAT OF RED OXIDE PRIMER (F5531FB).
K- COUPLING TO BE THOMAS SERIES 54 WITH STAINLS STL DISCS SIZE 312. BORE DRIVEN HALF 2.9355 DIA. W/ .750 x .375 K.W. - BORE DRIVING HALF 2.373 W/ .625 x .312 K.W.
L- FAN CASING TO BE CONTINUOUSLY SEAL WELDED TO MINIMIZE CORROSION. THIS INCLUDES ALL STIFFENERS AND ATTACHMENTS.

BRAUN/CONOCO- MONSANTO
LAKE CHARLES FEEDSTOCK VENTURE
TITLE OF EQUIPMENT - CRUDE HEATER
F.W.E.C ORDER Nº 5-16-991.
C/M AFE 6677- BRAUN ITEM Nº H-1101
BRAUN REQUIREMENT Nº 20

CUSTOMER PLEASE NOTE
1. FOR FAN FOUNDATION DESIGN CRITERIA REFER TO DWG 3333D64
2. W.E. CORP FURNISHES ONLY PARTS CALLED FOR ON THE B/M OF THIS DWG
3. W.E. CORP DOES NOT FURNISH ANY CONCRETE FOUNDATION, FOUNDATION BOLTS, OR SHIMS.
4. LEAVE FOUNDATION 1-2 IN LOW FOR PERMANENT STEEL SHIMS AND GROUT
5. PERMANENT STEEL SHIMS MUST BE INSTALLED ON TWO SIDES OF EACH FOUNDATION BOLT.
6. LEAVE .50 IN CLEARANCE AROUND ALL FOUNDATION BOLT TO ALLOW FOR ADJUSTMENT
7. PROVIDE .06 IN. TO .80 IN. THICK PERMANENT STEEL SHIMS BETWEEN BASE OF BEARINGS AND TOP OF PEDESTALS.
8. CUSTOMER MUST FOLLOW FAN INSTALLATION AND START-UP PROCEDURES PROVIDED IN INSTRUCTION BOOK

Westinghouse Electric Corporation
CONTRACT DRAWING 4054-B AIRFOIL FAN
SW/DSW ARR 7 CL-400 CW ROTA 45 TAL DISCH
8589D77

IP7_039046

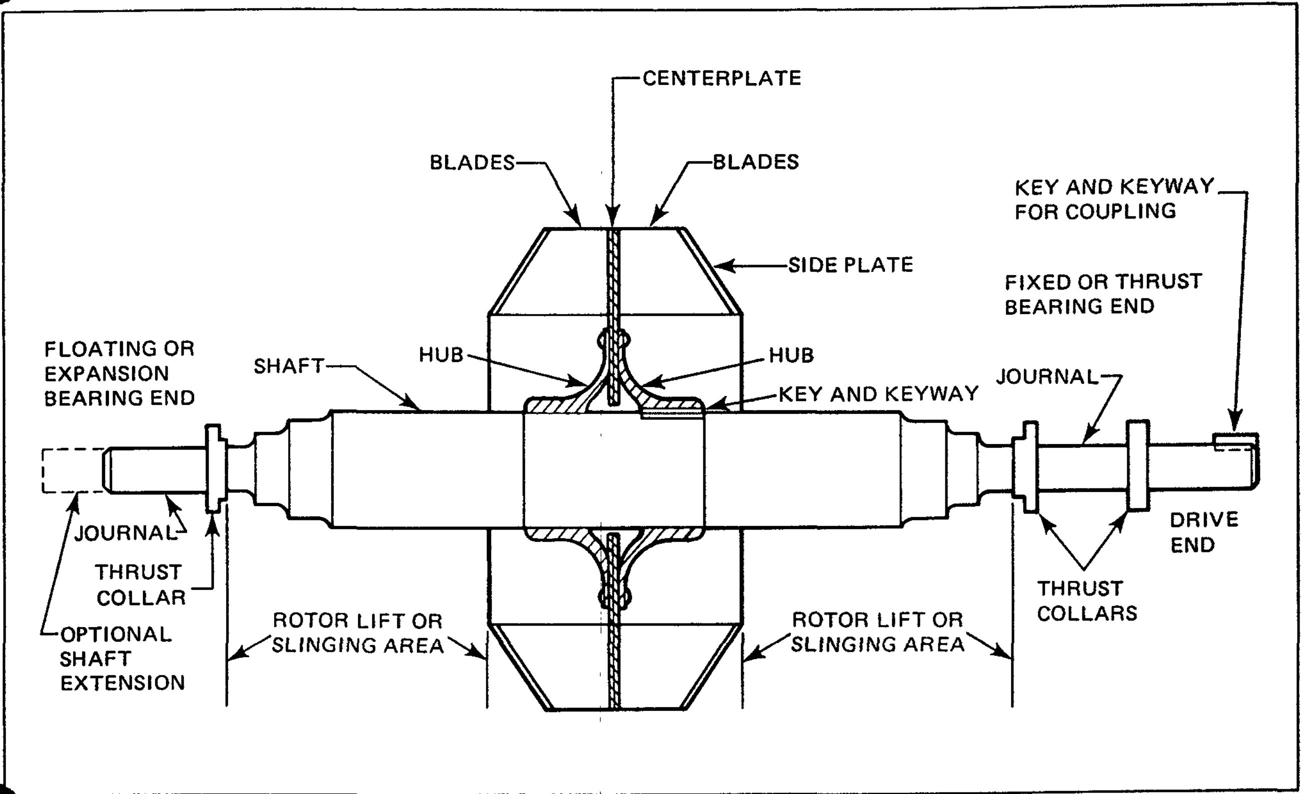


Fig. 5.7.1-2 *Typical Rotor Component Identification*

determined from the Main Drive end as specified by customer. See Figure 5.7.1-3 for Rotation and Wheel Blade Types.

6. Identify and move Fan Inlets or Vane and Inlet Assemblies to their respective Rotor ends.

- Inlets without Vanes are interchangeable and can be assembled to either side of Double Width-Double Inlet Fan.
- Vane and Inlet Assemblies are NON-interchangeable, and must be installed on the correct side of fan. Refer to Section 8 - Inlet Vane Controls, Preparation for Installation.

NOTE

Refer to Contract/Main Assembly Drawing for position of Vane Operating Lever. Vane Operating Lever must be in that approximate position when Vane and Inlet Assemblies are placed on Rotor.

7. Guide Inlets or Vane and Inlet Assemblies on Shaft. Place in approximate installation position; support securely to prevent damage to equipment or injury to personnel.

NOTE

Assure that all Non-Split Inlet Parts, such as Inlet Vane Control Seal Parts, are placed on Shaft in proper installation sequence.

8. Pressurized Air Shaft Seals Only

Refer to Contract/Main Assembly Drawing, if Pressurized Air Shaft Seal Assembly is listed in Bill of Material; refer to Section 11.3 for seal details. (If not listed, proceed to Step 9.)

Seal Bodies are to be placed on the Rotor in their respective position to their location on the fans.

CAUTION

SEAL BODIES HAVE MACHINED SURFACES AND MUST BE HANDLED WITH CARE.

Insert a minimum of four (4) wooden wedges between machined Seal Body inside diameter and Shaft outside diameter and secure into position.

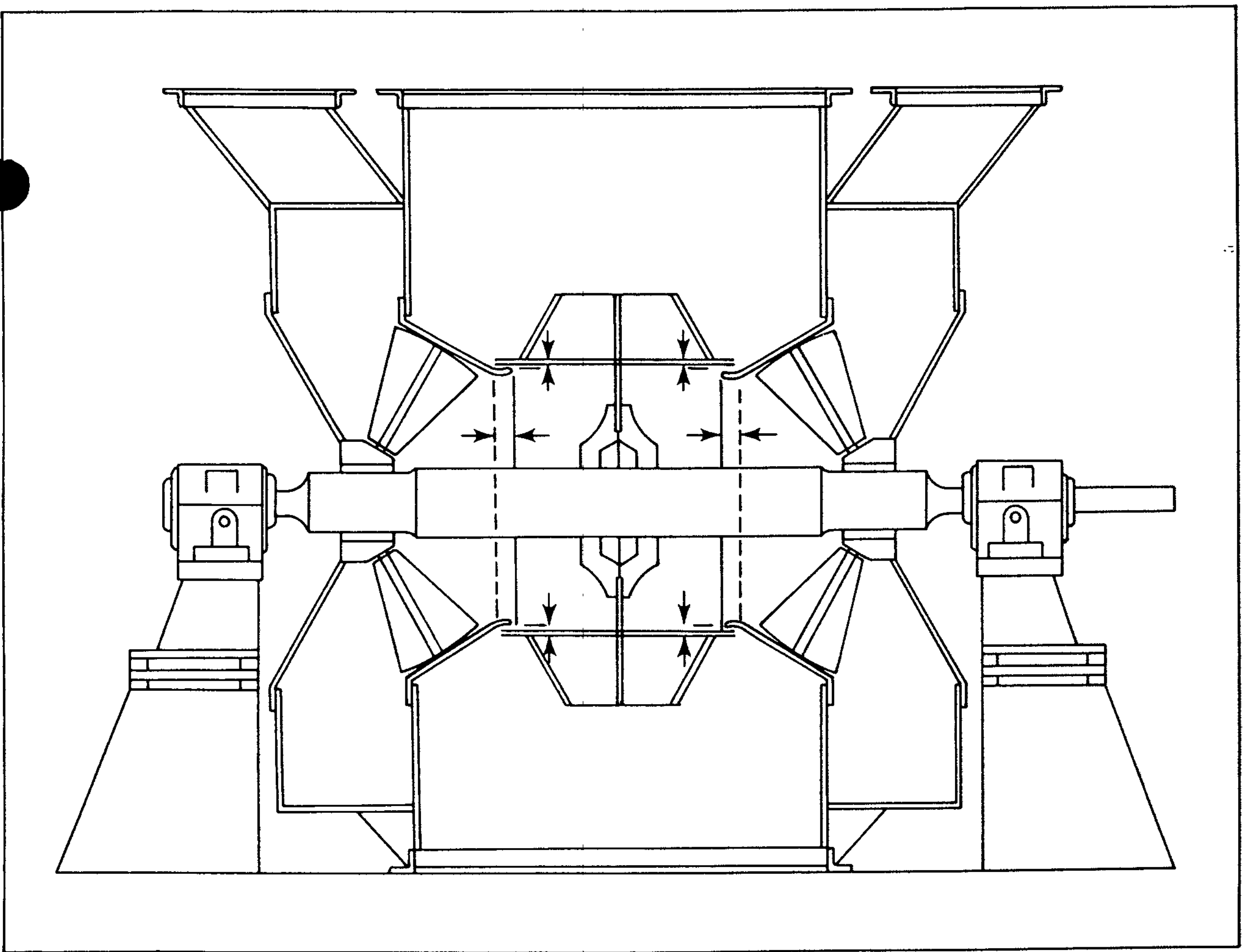
9. Install Bearings on Rotor. Refer to Section 6 - Bearings.

Check complete Housing/Inlet Box alignment in relation to Rotor and all center lines. Verify that Side Sheets are vertical. Correct as required.

2. INLET INSTALLATION AND ALIGNMENT

NOTE

- **Rotor must be leveled and aligned to Drive Train Centerline and at proper elevation.**
- **Inlets must be aligned to Wheel as shown in the Wheel and Inlet Relationship detail on Contract/Main Assembly Drawing. See Figure 5.5.3-14 for typical Wheel and Inlet relationship details. Figure 5.5.3-13 shows the Rotor and Housing/Inlet Box in cross section.**
- **The Wheel-to-Inlet relationship detail on Contract Drawing will show the cold or ambient temperature setting. Fans for elevated temperature applications will have a clearance at the top of Wheel greater than at the bottom in the cold condition. The Wheel and Inlet will come into alignment at operating temperature.**
- **Vane and Inlet Assemblies must be positioned with the Operating Lever and Ring Assembly in the position Indicated on Contract/Main Assembly Drawing.**
- **Final axial adjustment on Series 2300 and 2400 Fans must be made by loosening machine screws holding the Sliding Seal Ring to the Fabricated Steel Inlet, and sliding the Seal Ring to the correct position. Tighten all machine screws securely after alignment.**



g. 5.5.4-13 *Typical Cross Sectional View of Fan*

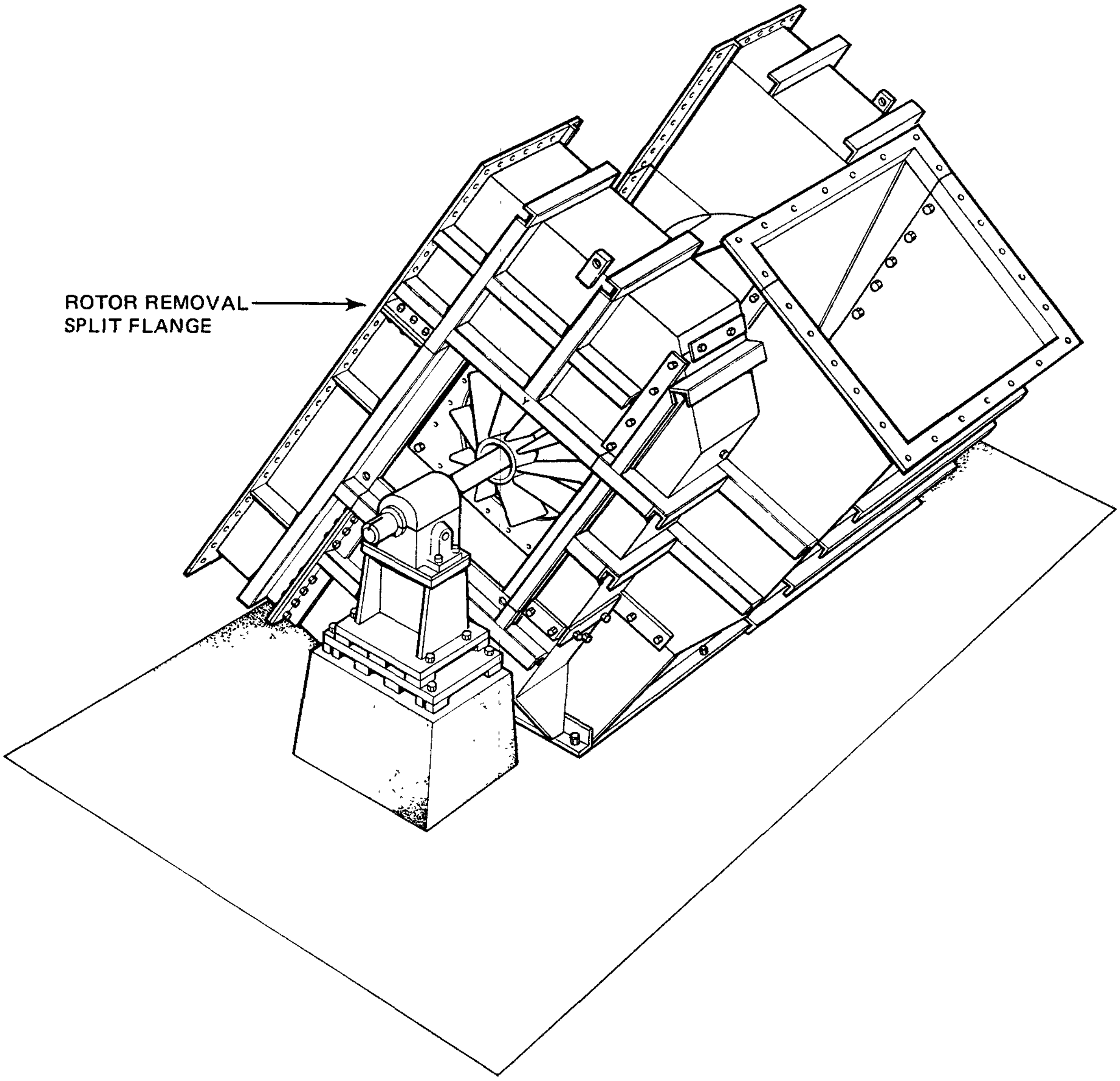


Fig. 5.5.4-11 *Housing Construction - Type 4 Tenth Erection Stage*

N. Place gasket material (if required) on Rotor Removal Split Flanges, and tie or tape temporarily. Lift Rotor Removal section in position. Use drift pins to align bolt holes in the mating angle iron flanges.

If sections do not line up, check assembly for misaligned sections installed previously. The Housing/Box sections were factory-mated and match marked. If all sections are correctly aligned they will mate properly. Correct any misaligned sections.

P. Install and tighten all bolts and nuts in Rotor Removal Section Flanges. Refer to Torque Values in Section 5.10.

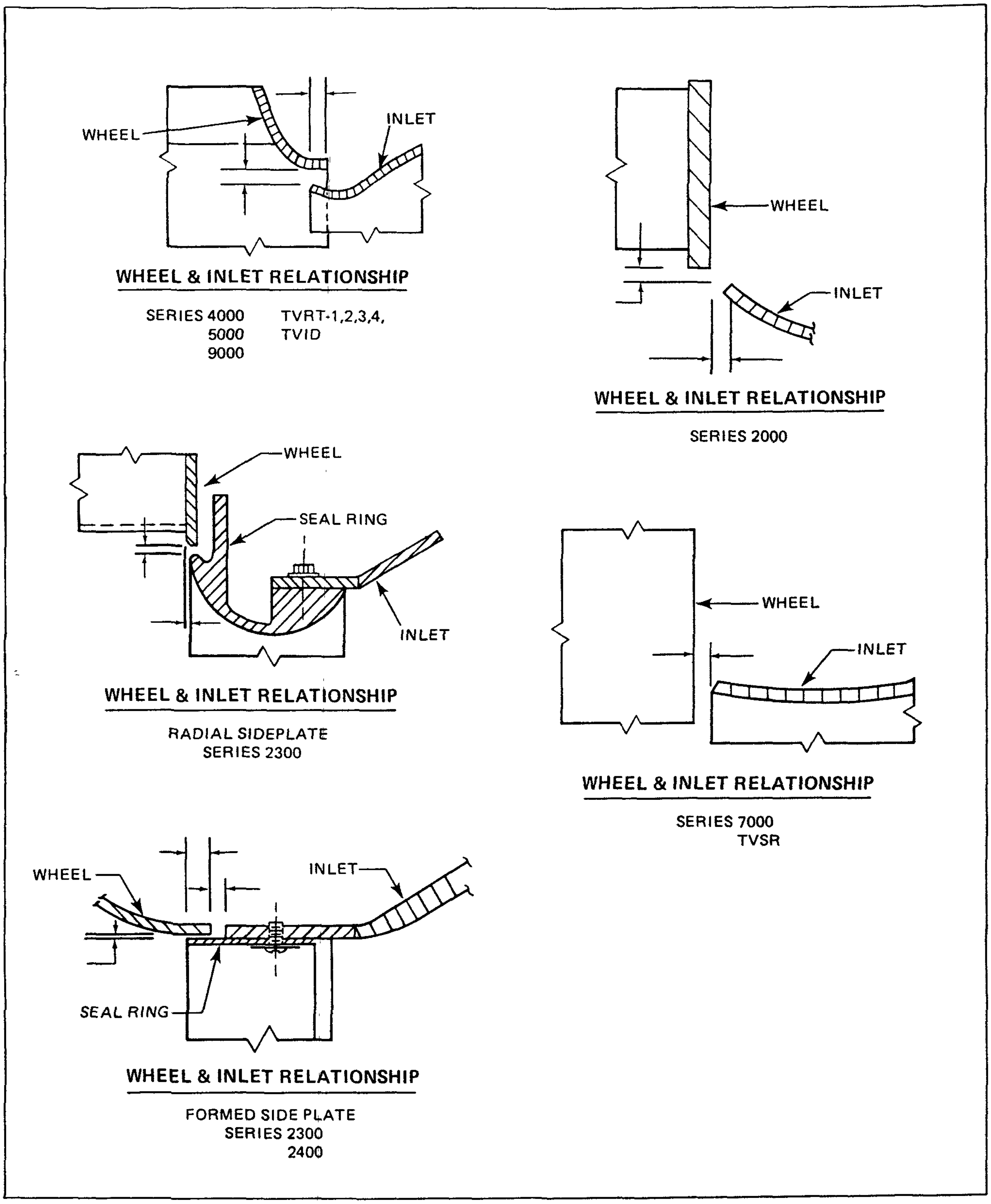


Fig. 5.5.4-14 *Typical Wheel-Inlet Relationship Details*

each rotating shaft and complying with the requirements of the 1448
State of Utah shall be furnished for each coupling of each fan. 1449
The coupling guard shall be provided with a removable plate or 1450
other equivalent means for inspection and oiling.

21.8 Sole Plates: Steel sole plates for each fan 1453
bearing pedestal shall be provided. Bearing pedestals shall be 1454
bolted to the IPA foundation by means of anchor bolts extending
through the sole plates.

21.9 Primary Air Fans: Primary air fans shall be 1457
provided as necessary for proper boiler operation. With one 1458
primary fan out of service the remaining fans shall be capable
of providing sufficient primary air to permit boiler operation 1459
up to not less than 60 percent of Maximum Capacity with each of
the specified coals. Test block performance of each fan with 1460
inlet boxes and silencers shall be greater than its expected
operating performance by not less than 25 percent for weight 1461
flow, not less than 50 percent for static discharge pressure,
and not less than 25F for temperature at the plant's elevation 1462
for each of the specified coals.

The primary air fans shall be of the full shrouded 1464
type with backwardly curved airfoil blades and shall be double 1465
width, double inlet. The fans shall have inlet boxes complete 1466
with silencers. An annubar and 2 thermocouples shall be 1467
provided in each inlet box for measuring the primary air flow.

The bearings shall be of the self-aligning, split 1469
sleeve type capable of withstanding high thrust loads due to any 1470
unbalanced forces on the fan wheel. The bearings shall be 1471
cooled with oil from the lubricating oil package specified
herein. The bearing lubrication system shall be of the flow- 1472
through flood type.

An independent lubricating oil package shall be 1474
furnished for each fan and motor unit. Each lubricating oil 1475
package shall be capable of supplying the required amount of
lubricating oil to the fan and motor bearings at test block 1476
conditions.

Each package shall be complete with dual full-capacity 1478
water-to-oil heat exchangers, dual full-capacity oil filters, 1479
and dual full-capacity pumps and motors. The filter shall be 1480
cleanable or replaceable without interrupting the oil flow. The 1481
complete package shall be factory-wired, assembled, and mounted
on a common base. The packaged system shall be arranged and 1482
wired such that either pump may serve as the main supply pump
while the other pump serves as the standby pump. 1483

The heat exchangers shall have 90-10 copper-nickel 1485
tubes and bronze tube sheets and shall be sized for 105F cooling 1486
water.

The oil reservoir tank shall be coated internally with 1488
Rustban or other commercially available rust inhibitor. The 1490
reservoir shall be complete with dual 480-volt, 3-phase, 60-
hertz, electric heaters and an oil level site gage. The heaters 1492
shall be complete with in-line full voltage adjustable
thermostatic or combination thermostat and contactor controls 1493
capable of interrupting the heater circuits. The heaters shall 1494
automatically maintain the temperature of the oil in the
reservoir at the selected temperature when the fan is idle and 1495
the ambient temperature is as low as -30F. The entire oil 1496
reservoir electric heater circuit shall be factory-wired to
terminals in the common junction box so that only connection of 1497
480-volt supply is necessary to complete the heater circuit. A 1498
metal barrier shall be provided in the junction box between
terminals for the 480-volt supply and terminals for the control 1499
circuits.

Each lubrication package shall be furnished complete 1501
with all instrumentation and control devices required for 1502
monitoring and control. All devices shall be mounted, piped, 1503
and wired as part of the package to comprise a complete system.
The devices shall include, but not be limited to, the following: 1504

(a) One low-pressure switch for alarm. 1506

(b) One high temperature switch for alarm. 1508

(c) One permissive start pressure switch for fan 1510
motor.

(d) One low-pressure trip switch. 1512

(e) One low-pressure start switch for each lube- 1514
oil pump.

(f) One low temperature switch for alarm. 1516

(g) Two standby-auto-run pump selector switches 1518
with eight contacts.

(h) One level switch for oil reservoir low level 1520
alarm.

(i) Two full-capacity immersion type heaters or a 1522
single heater that will be serviceable with the primary air fan 1523
operating. A spare heating element shall be furnished. 1524

(j) Thermostats. 1526

(k) Flow indicators. 1528

(l) Pressure gages. 1530

(m) Temperature gages. 1532

(n) Level gages. 1534

Pump selector switch developments will be provided to 1536
the Contractor by the IPA.

Motor starters for pumps and auxiliary relays required 1538
for control shall be furnished.

At the driven end of the fan shaft, suitable thrust 1540
bearings shall be provided to locate the shaft longitudinally 1541
and to compensate for end thrust under any condition of
operation.

Connector lead type resistance temperature detectors 1543
(RTD) shall be furnished and installed on each bearing.

Pillow blocks shall be carried on pedestals 1545
independent of the fans. Should cast-iron pillow blocks be 1546
furnished, these shall be of good quality iron, free from
defects, imperfections, blow holes, or cracks. No patching or 1548
welding will be permitted unless specifically approved by the
Engineer.

Bearing pedestals shall be of heavy, rigid, steel 1550
plate construction, mounted on sole plates to facilitate removal 1551
of the motor bearings without disturbing the alignment of the
fans or driving motors.

The Contractor shall furnish asbestos gaskets or 1553
pressure sensitive gasket material for all bolted flange joints 1554
and manhole joints of the fan housings.

The gaskets for the flange joints at the discharge 1556
connections to the duct work shall be such that after the 1557
connections have been securely bolted and the gaskets compressed
the gasket thickness shall be 3/16 inch or greater. 1558

All exposed parts of the shaft shall be polished and 1560
the wheel hubs, fan blades, and shrouds shall be smooth and 1561
uniform and of such contour as to offer the least resistance to
the flow of gases or air. Castings shall be carefully dressed 1562
down and filled.

21.10 Blowers: Contractor shall furnish all blowers 1565
required to seal, cool, vent, or purge components furnished by
the Contractor except components otherwise specified as being 1566
furnished with the required air by the IPA. Blowers shall 1567
include spare capacity as required to maintain boiler capability
from start-up to Maximum Capacity upon failure of any one blower 1568
or drive.

22. Control Equipment: Except for the equipment 1571
under Subarticle 22.6 of this Division, the following control 1572
equipment shall be furnished:

22.1 Dampers: The maximum allowable deflection of 1575
damper blades under all service conditions shall not exceed
1/360 of the span. Control dampers used for throttling service 1576
shall be of the counter-rotating type.

All damper bearings shall be so mounted as to prevent 1578
binding of the shafts because of thermal expansion or 1579
misalignment.

Each damper operating shaft shall be provided with 1581
keyways in each end and shall be supported in easily lubricated 1582
anti-friction bearings. All damper shafts shall show damper 1583
position and shall have open and closed positions clearly
indicated.

Each damper blade and inlet vane shall be continuously 1585
overlapping with no fixed division plates, and shall be of the 1586
design that is rigid, and that will operate free from flutter or
vibration during all modes of operation. The Contractor shall 1588
provide a means of lubrication to individual inlet vanes to
ensure free and trouble free operation of the inlet vanes. 1589

If the damper and inlet vane shafts operate in 1591
sleeves, provisions shall be made to eliminate infiltration of 1592
dust and moisture between the shaft and the sleeve. The 1593
Contractor shall provide a positive means of determining the
position of the dampers.

Torque requirements for damper and inlet vane 1595
mechanisms shall be held to as low a value as practicable.

	Hybrid		Constant Pressure				
Load..........percent of Maximum Capacity	25	50	25	50	75	100	105
Air flow rates:							
Theoretical air for combustion..Mlb/hr	1641	3162	1636	3104	4463	5685	6047
Total air for combustion, including excess air..........Mlb/hr	2257	3952	2248	3880	5266	6538	6954
Air from primary air fans.......Mlb/hr	550.0	1038.5	541.0	1037.5	1293.0	1552.0	1601.0
Tempering air...................Mlb/hr	98.2	370.3	101.1	376.9	425.2	485.8	484.6
Air to primary air heaters......Mlb/hr	451.8	668.2	439.9	660.6	867.8	1066.2	1116.4
Air from primary air heaters, exclusive of tempering air....Mlb/hr	300.8	528.2	293.9	521.6	713.8	903.2	947.4
Air from F. D. fans.............Mlb/hr	2066	3305.5	2062	3231.5	4443	5540	5935
Sealing air*...................Mlb/hr	22	55	22	55	66	77	77*
Air to secondary air heaters....Mlb/hr	2066	3305.5	2062	3231.5	4443	5540	5935
Air from secondary air heaters.......................Mlb/hr	1880	3108.5	1875.0	3036.5	4193	5226	5599
Air pressures:							
Air from primary air fans.in. of water	30.1	26.0	29.4	25.7	30.1	34.2	36.3
Air to primary air heaters.................in. of water	29.7	24.3	29.0	24.0	28.3	31.5	33.4
Air from primary air heaters.................in. of water	29.5	23.8	28.8	23.5	27.5	30.2	32.0
Air to secondary air heaters.................in. of water	1.8	1.3	1.7	1.2	3.6	5.2	6.0
Air from secondary air heaters.................in. of water	1.4	0.5	1.3	0.4	2.2	3.2	3.7
Air at windbox............in. of water	1.0	-0.2	0.9	-0.3	0.8	1.1	1.4

*Sealing air is taken from primary tempering air.

	Hybrid		Constant Pressure				
Load.........percent of Maximum Capacity	25	50	25	50	75	100	105
Air temperatures:							
Minimum ambient temperature not requiring preheating..............F	84	60	94	60	50	37	34
Air to secondary air heater.........F	89**	62	99 **	62	63	64	65
Air from secondary air heater........F	534	590	507	590	615	635	645
Air from primary air heater, excluding tempering air...........F	511	562	510	561	567	564	569
Air to pulverizers...................F	420	380	415	375	402	412	420
Air to Primary AH..................F	75	75	75	75	75	77	78
Flue gas flow rates:							
Leaving furnace................Mlb/hr	2492	4343	2484	4264	5787	7184	7643
Leaving economizer.............Mlb/hr	2509	4375	2500	4295	5832	7241	7703
Entering primary air heaters....Mlb/hr	363.9	682	439.8	670	839	972	991
Entering secondary air heaters.....................Mlb/hr	2145.1	3693	2060.2	3625	4993	6269	6712
Leaving primary air heaters.....Mlb/hr	514.9	822	585.8	809	993	1135	1160
Leaving secondary air heaters...Mlb/hr	2331.1	3890	2247.2	3820	5243	6583	7048
Maximum velocity of gases in superheater net free area........ft/sec	11	22	11	22	26	45	51
Maximum velocity of gases in reheater net free area...........ft/sec	23	39	28	40	50	46	47
Maximum velocity of gases in economizer net free area.........ft/sec	6	14	4	13	22	36.2	41

**Steam Coil Heating used to heat the air from a temperature of 62 F.

	Hybrid		Constant Pressure				
Load.........percent of Maximum Capacity	25	50	25	50	75	100	105
Flue gas pressures:							
In furnace...............in. of water	0	0	0	0	0	0	0
Leaving furnace..........in. of water	-0.0	-0.0	-0.0	-0.0	-0.1	-0.1	-0.1
Entering economizer.......in. of water							
Leaving economizer........in. of water	-0.9	-1.5	-1.0	-1.5	-2.4	-2.7	-3.1
Entering primary air heater.................in. of water	-1.6	-2.4	-1.7	-2.4	-3.7	-4.1	-4.6
Entering secondary air heater.................in. of water	-1.6	-2.4	-1.7	-2.4	-3.7	-4.1	-4.6
Leaving primary air heater.................in. of water	-2.0	-3.4	-2.2	-3.4	-5.2	-6.0	-6.4
Leaving secondary air heater.................in. of water	-2.2	-3.9	-2.3	-3.9	-6.2	-7.9	-8.8
Flue gas temperatures:							
Leaving furnace (12" SS)....................F	1190	1480	1170	1470	1650	1780	1815
Entering superheater (24" SS)..........F	1375	1740	1360	1725	1940	2075	2115
Entering reheater (PENDANT)..........F	1092	1330	1064	1325	1480	1590	1625
Entering economizer.................F	589	710	653	723	773	828	848
Leaving economizer..................F	581	654	554	653	694	724	740
Entering primary air heater.........F	581	654	554	653	694	724	740
Entering secondary air heater.......F	581	654	554	653	694	724	740
Leaving primary air heater (Incl LKG) F	200	278	237	277	280	280/280	280
Leaving secondary air heater (Incl LKG) F	211	237	203	237	263	276/280	280
						7 Mills/8 Mills	

	Hybrid		Constant Pressure				
Load.........percent of Maximum Capacity	25	50	25	50	75	100	105
Excess air:							
Leaving furnace...............percent	42	27	42	27	19.3	16	16
Leaving economizer............percent	43	28	43	28	20.3	17	17
Leaving primary air heater.....percent	64*	39*	63*	39*	29*	26*	26*
Leaving secondary air heater...percent							

* Pri. and Sec. AH's combined

	Hybrid 25	Hybrid 50	CP 25	CP 50	CP 75	CP 100	CP 105
Emission concentrations:							
Dust loading lvg. Ah's-grains/SCF	4.73	5.52	4.74	5.51	5.88	6.04	6.04
Dust loading leaving economizer...............grains/SCF	5.37	5.94	5.37	5.94	6.29	6.44	6.44
SO_2 concentration leaving economizer.......................ppm	8.7	9.3	8.5	9.3	9.9	10.1	10.1
NO_x leaving economizer........lb/MMBtu	---	---	Maximum of 0.55 lb/MKB		---	---	---
NO_x leaving air heaters.......lb/MMBtu	Maximum of 0.55 lb/MKB (corrected for leakage) across load range.						

Heat absorbed by Boiler:

**H_2 and H_2O combined

	Hybrid 25	Hybrid 50	CP 25	CP 50	CP 75	CP 100	CP 105
Losses:							
Hydrogen in fuel............percent	4.93	5.10	4.89	5.10	5.13	5.15	5.16**
Moisture in fuel............percent	---	---	---	---	---	---	---
Moisture in air.............percent	.05	0.06	0.05	0.06	.06	0.07	0.07
Dry gas.....................percent	3.63	4.57	3.42	4.57	4.74	4.84	4.98
Combustible in refuse........percent	0.20	0.20	0.20	0.20	0.20	0.20	0.20
Radiation...................percent	0.57	0.30	0.58	0.30	0.21	0.17	0.15
Unaccount & mfg margin.......percent	1	1	1	1	1	1	1
Total heat absorbed...........percent	10.38	11.23	10.14	11.23	11.34	11.43	11.56
Boiler Efficiency (Per ASME PTC 4.1 abbrev. test method)	89.63	88.78	89.87	88.78	88.67	88.58	88.45
Fuel efficiency (with heat credits)	89.72	88.88	89.96	88.87	88.76	88.67	88.54

	Hybrid		Constant Pressure				
Load.........percent of Maximum Capacity	25	50	25	50	75	100	105
Power input to motors:							
Number of pulverizers operating..........................	2	5	2	5	6	7	7
Pulverizer motors..................kW	886	1904	876	1886	2494	3048	3152
Number of primary air fans operating..........................	1	1	1	1	2	2	2
Primary air fan motors.............kW	1098(.920)	1341(.935)	1058(.920)	1325(.935)	2132(.933)	2447(.934)	2549(.935)▲
Number of coal feeders operating......	2	5	2	5	6	7	7
Coal feeder motors.................kW	3	8	3	8	10	11	11*
Number of boiler circulation pumps operating.....................	N/A	N/A	N/A	N/A	N/A	N/A	N/A
Boiler ciculation pump motors.......kW	N/A	N/A	N/A	N/A	N/A	N/A	N/A
All other motors...................kW	103	128	103	128	133	138	138**

* Based on Merrick Feeders
** Seal Air Fans and Regenerative Air Heaters
(Two 24-1/2 VI 44 Primary and Two 33-1/2 VI 64 Secondary's)
▲ Westinghouse Two Speed Fans (23120D) with Two Speed Motors.
( ) Indicates Expected Motor Efficiency

E.O.M. INSTRUCTIONS

GENERAL DATA

NUMBER AND SIZE FANS	(2 x 2) FAF 37.5/18-1
APPLICATION	FD Horizontal - Indoor Design
INSTALLATION YEAR	1984/1985
FAN NUMBER	1A 1B 2A 2B
MACHINE NUMBER	1124 1125
FLOW - T.B. CFM	1,154,700 ACFM
TEMPERATURE ^{O}F	110
PRESSURE INS. WG.	21.28
FAN INPUT H.P.	4305
FAN SPEED RPM	880/720
MOTOR SIZE H.P.	6500/2750
MOTOR SPEED RPM	880/720
PLANT ELEVATION FT.	4676'
FAN CHARACTERISTIC CURVES	BA-560 Page 5-4A Actual Curves in Section 9